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MEETING IN GERMANY

For the more complete arranging of the joint meeting of the Society with the Verein deutscher Ingenieure, to be held in Leipzig, June 23-25, 1913, Dr. Conrad Matschoss, Dozent of the Royal Polytechnic High School of Berlin, is now in America representing the German society, and tentative plans will soon be announced. From the details thus far known it can be foretold that a most unusual opportunity will be afforded to visit the various German industries and to observe the progress made along engineering lines in that country. The textile district around Leipzig, the machine tool and various other industries at Düsseldorf, the Allegemeine Electricitäts Gesellschaft at Berlin, the largest manufacturers of electrical apparatus in the world, Siemens Schuckert, Ludwig Loewe, who entertained members of the Society during the visit of 1889, and the chemical works in Frankfort, are being considered as points of interest to be included in the program. Special opportunities will also be provided for a comprehensive study of the great German Museum of the Technical Arts and Industries at Munich. There will also be a trip on the Rhine and receptions in several of the cities, and visits to the various shipyards and to the art galleries in Dresden and other cities. In addition to these official excursions,

invitations are being received by the Society from a number of prominent German firms which offer their hospitality to our members and show the great interest which is felt in the trip.

The Committee of Arrangements, Col. E. D. Meier, Chairman, is assured of adequate steamer accomodations, affording the maximum of comfort with the minimum of expense, provided the party is large enough. It is desired to charter a steamer, which will make possible the gratifying of all tastes. Guests will be welcomed and may be included in the party.

In addition to his mission in connection with the Joint Meeting, Dr. Matschoss is charged by the Verein deutscher Ingenieure to make a tour of America to study our educational methods and to report on these in connection with the part they are taking in the modification of the curriculum in Germany so as to produce technical graduates who are more immediately available in the shop. The Council and the Committee of Arrangements for the Joint Meeting have in charge an itinerary for Dr. Matschoss which will enable him to secure the information he seeks, and members of the Society in the various cities which he may visit are urged to coöperate by placing facilities for securing such data at his disposal.

CURRENT AFFAIRS OF THE SOCIETY

The opportunities for serving the profession of engineering were never greater than they are at the present time. The Society is seeking to enlist in this service the interest of the most eminent engineers in every branch to contribute of their knowledge either papers or discussions, and members should extend to men of attainment invitations to become identified with the Society.

All suggestions along such lines are being actively followed up by the Committee on Increase of Membership, composed of members of the Council and other members of the Society, and the coöperation of the membership generally is encouraged. Accompanying this increased activity, however, is the increasing strictness of the Membership Committee in its interpretation of the requirements for admission. The membership can greatly assist by explaining to applicants the essential requirements for admission and the necessary care which must at all times be enforced.

VISIT OF DR. UNWIN

Dr. William Cawthorne Unwin, Honorary Member of the Society and President of the Institution of Civil Engineers of Great Britain, is expected to arrive in this country the latter part of August, and will be welcomed by many members of the Society.

WILLIAM MARSH RICE INSTITUTE

The Society will be officially represented at the opening of the William Marsh Rice Institute in Houston, Texas, on October 10-12, by its President, Dr. Alexander Crombie Humphreys, president of the Stevens Institute of Technology. The new institute is destined to become one of the principal institutions of the South and men of letters and science from all countries of the world will gather to celebrate its opening.

INTERNATIONAL CONGRESS FOR TESTING MATERIALS

The members of the Society and ladies are cordially invited to attend the joint reception given in honor of the delegates to the International Congress for Testing Materials by the American Society for Testing Materials and the three founder societies, in the Engineering Societies Building, New York, on the evening of September 2. No further notice of this meeting will be made to the membership, but it is anticipated that a large audience will assemble to greet the delegates.

CALVIN W. RICE, *Secretary*

MEETING OF THE SUB-COMMITTEES ON THE INDUSTRIES

A meeting of the newly appointed sub-committees of the Committee on Meetings, including committees on Textiles, Administration, Cement Manufacture, Machine Shop Practice, Iron and Steel, Air Machinery, Railroads, Hoisting and Conveying, Industrial Building, and Fire Protection, was held in the Engineering Societies Building on July 29, and was very largely attended. Dr. Alex. C. Humphreys, President, and Calvin W. Rice, Secretary, were present, and the Committee on Meetings was represented by H. deB. Parsons and Dr. Charles E. Lucke, Chairman.

Dr. Lucke, who acted as Chairman of the meeting, outlined its

general purpose, stating that the Society was now making an experiment, the result of which rested with the sub-committees and the individual members. In addition to the meetings held for the discussion of strictly professional matters, directed for the benefit of the members, it is proposed to bring out the broad problems common to many or all of the different industries, applying to their solution the methods adopted by the most successful. The emphasis will, of course, be laid on those problems which properly fall within the province of engineers, rather than of traders, lawyers, or financiers.

For the working out of this experiment, groups of men have been appointed, known as committees, charged with the responsibility of bringing before the Society both the problems and their successful solution, each group being invited to coöperate with the other in any presentations which may be made. Specialists on the various subjects have been chosen and the subjects will be treated in such a way as to be helpful to all. To conform with the professional ideals of the Society, these presentations must represent scientific advance and at the same time be of practical benefit. If possible whole sessions of the Annual Meeting will be assigned to the sub-committees, for a complete discussion of the problems presented.

A general discussion followed Dr. Lucke's remarks.

The Committee on Cement Manufacture, F. W. Kelley, chairman, has already chosen as its leading topic for the Cement Session of the Annual Meeting, Depreciation and Obsolescence, and a letter has been issued to other sub-committees requesting statistics as to the obsolescence of all types of apparatus. The following are specifically mentioned: electric generators, steam engines, turbines and auxiliaries; piping, pipe fittings, steam boilers, boiler auxiliaries, stokers and coal handling equipment, electric lines and line apparatus, electric motors, lineshafts and pulleys. A second topic which has been suggested, Prevention of Accidents to Employees, also demands coöperation from other committees, and data have been asked for upon the protection of gears, pulleys, shafting, electric wiring, overhead runways and platforms, and in general the methods of securing the attention and coöperation of the workers themselves in conjunction with mechanical means for preventing accidents to them. An invitation to coöperate on these topics is extended to the membership at large.

THE PRINCIPLES OF VALUING PROPERTY

WITH SPECIAL REFERENCE TO INDUSTRIAL ENTERPRISES

By Henry K. Rowell

ABSTRACT OF PAPER

The purpose of the paper is to bring out some of the fundamental principles involved in the valuation of property. The distinction is defined between the word value, and a valuable consideration, and the several purposes for which value is to be determined, each of which may vary from the other and still be within the truth. The commercial value is the one most essential to an industry and should be the determining factor for purposes of tax, insurance or sale.

No valuation can be properly ascertained without accounting for the depreciation.

Capitalization is a delicate and intricate question to deal with. The author's experience is that most if not all writers skip over this subject lightly, and he has yet to find a real logical discussion of the subject.

In all mechanical and industrial processes the same laws of growth and development are as potent as are the laws of evolution in the natural growth and development of the physical world.

THE PRINCIPLES OF VALUING PROPERTY

WITH SPECIAL REFERENCE TO INDUSTRIAL ENTERPRISES

By Henry K. Rowell, Boston, Mass.

Member of the Society

In general every industry should be revalued at stated periods. The laws of some states require a revaluation every five years, or for purposes of assessment or insurance every seven years. In this paper the leading principles laid down for defining and ascertaining the value of property for various purposes are concisely expressed.

2 In discussing the several kinds of values, reference is made to a number of cases handed down from the Supreme Courts of several states and the United States, and some of the cases establishing the precedent upon which the rules of law are founded may be found in Judson's Taxation. In some cases the language used is that of the reported cases and in others of the textbook. The matter contained in the discussions of depreciation and capitalization is out of the author's own experience, with a brief reference to the report of the Tariff Board on the worsted and woolen industry.

DEFINITION OF TERMS

3 Following is a definition of the terms used, or referred to:

a *Cost of plant* includes the cost of the materials, labor, professional services required to produce the physical property, interest and insurance during construction, and the amount required to put the plant in operation.

b *Cost of operating* includes labor, supplies and incidentals necessary to manufacture the product, but does not include the stock in process, repairs and maintenance.

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c *Fixed charges* includes taxes, insurance, interest on current liabilities, and bonds.

d *Overhead charges* includes oversight, office and counting room expenses, advertising and selling.

e *Depreciation annuity*, the sum set aside as a sinking fund to replace the plant.

f *Annuity*, the sum set aside to discharge the funded debt at the end of a stated period.

g *Amortization*, the annuity set aside as a sinking fund to redeem the capital invested at the end of a stated period. The term refers particularly to the death of an industry from its exhaustion, as a mine or quarry. It is not depreciation, although analogous to it. Depreciation infers a continuance by renewals to the tangible property. The term amortization is not again referred to in the following discussions.

VALUE OF PROPERTY

4 In determining the value of property all the factors enumerated enter into the problem, and each must be given its proper amount of consideration and correct application to bear the test of logical reasoning. Every case of appraisal should be approached with the idea that the results may have to stand the test of the court.

5 Value does not mean cost, nor does it necessarily mean the purchase nor the selling price. It means the desirability or worth of one thing as compared with that of another, with reference to its usefulness. Value as defined by law is measured in terms of money.

6 A valuable consideration, however, within the meaning of the law may consist either in some right, interest, profit or benefit accruing to one party, or some forbearance, detriment, loss or responsibility given, suffered or undertaken by the other. It is under this definition that a court of equity would compel the specific performance of a contract where damages would not give adequate compensation as in a case where the thing contracted for is specific and its counterpart can not be purchased in the open market.

7 The rule of law would be to place a value upon the loss or non-performance of the promise and award the damages in a payment of money. Equity follows the law wherever possible,

and in a suit for damages where compensation can be given by the payment of a sum of money determined by the value of the property in question, equity would not grant specific performance, nor what is termed a payment in kind.

8 The theory of the law is that money as the universally recognized medium of exchange and measure of value, is always acceptable and leaves the one who rightfully possesses it free to purchase what he wishes to satisfy his requirements or pleasure. It is very rare, indeed, that a demand or pleasure can not be satisfied by the purchase of something else.

9 Another reason for preferring a payment of damages in money is that upon the payment of money litigation ceases; whereas, if specific performance, or payment in kind were granted, the courts would have the difficulty of enforcing the specific performance, or payment in kind.

10 Payment in kind, in the case of water rights, has been allowed in England, by act of parliament.

TAX VALUE

11 Tax value is defined by the courts and stated by the legislators of the several states to be the true or fair cash value of the property at the time of assessment. The fair cash value is the price which a willing and voluntary seller will accept from a willing purchaser who has the money to pay at the time of sale. In the assessments of taxes the rates will vary from nothing in the case of exemption to over $20 per thousand. Some localities may have a low rate of valuation with a low rate of tax, others may have both a high rate of valuation with a high rate of tax, while others may have a moderate rate for both valuation and tax. There is no violation of the law as long as the rates are uniform and equal. The United States Supreme Court laid down the rule that uniform and equal refers to geographical uniformity where the rates operate generally throughout the United States, and uniformly wherever the subject of the tax is found. The court has declined to define the word equal as no rule can be formulated to cover every case, but upon the general question it is said that equal protection of the law means that no person or class of persons shall be denied the same protection of the laws enjoyed by other persons or other classes of persons in the same place and in like circumstances.

12 Some of the states hold that equal and uniform mean in-

trinsic equality and uniformity, which is not apparent equality or uniformity. The Supreme Court of Massachusetts said that a fair cash value is irrespective of value placed by the assessors upon similar property in the same city, the question being whether it had been valued at more or less than its fair cash value, and not relatively more or less than similar property of other persons.

13 The amount of taxes assessed is usually taken from about two-thirds to three-quarters of the full value of the property, but this is not always the case. While making an appraisal recently the author found that the rule of the locality was to collect the amount of tax upon 80 per cent of the full value, while the tax rate was $18 per thousand upon a fair valuation.

14 It is obviously immaterial what the basis of valuation is if it is uniform as to all property within the territory, or as to the class of subjects upon which the tax is laid. This is recognized in the requirements of state constitutions that taxation shall be uniform upon the same class of subjects within the territorial limits of the authority imposing it. Thus, if all the property in the state were valued on the same basis it would be immaterial to the individual tax payer whether he paid 1 per cent on a valuation of $1, or 2 per cent on a valuation of 50 cents. If there was no general property tax levied by the state, based upon valuation, it would make no difference whether property in one town or county was valued on a higher basis than property in another. But within the territory wherein the tax is levied, inequality of taxation results as certainly from inequality of valuation as from inequality in tax rate. The failure to recognize this fundamental principle in taxation can result only in misleading comparisons.

15 It has been held by the courts that the evidence of the valuation of other properties is admissible as a possible assistance in determining the cash value of the property in question, when the cash value and not the proportionate value is the point in issue.

16 Stock market quotations of the company's securities may be considered because the stock and indebtedness represent the property. But they are not to be regarded as conclusive standards or tests of value. In the language of the United States Supreme Court such quotations represent the faith which a purchaser of stock in such a company has in the ability with which

the company will be managed, and in the capacity to make future earnings. Whether it is well or ill founded, is but a matter of opinion. While the value of the property is one of the material facts going to make up the market value of the stock, it is not the sole one. Speculation has been known to exercise a potent influence on the market price of stock. But assessors can use this as evidence of value.

17 Capitalized net earnings as a means to determine the fair cash value of public utility, or quasi public corporations, including the stock, franchise and tangible property, has been allowed by the court. The basis of the valuation was taken as the net earnings for the year, which did not appear to be an exceptional one, and capitalized at 6 per cent; this ratio being adopted in a special case, is the rate adopted in states where assessments were made upon the basis of net earnings.

18 The court also said that this rate was less than some advanced advocates of municipal ownership were willing to guarantee to investors in securities of this character, but it was deemed justifiable by the considerations that attended the real investor's purchase of stock. To the value of the stock and franchises determined by capitalization of net earnings was added the value of the tangible property. In a case where there were no net earnings, but an apparent loss, the court said there was no basis upon which to assess the stock above the value of the tangible property.

INSURANCE VALUE

19 In cases of personal property the measure of damages for loss of the insured property is its market value at the time and place where the loss occurred, within the amount named in the policy. In the case of damage to an insured building, the rule is indemnification to the owner, not exceeding the sum insured. The question is not what someone would have paid for the building, but what amount will indemnify the insured for the loss sustained.

20 Where goods are insured the insurer is bound to pay their value at the time of loss, where damaged only he is bound to pay the difference between the value in the sound and damaged condition. Market value, and not local or peculiar value must control.

21 In the case of the destruction of a building, where the policy

is not valued, the damage would be the value of the property as it stood upon the ground the day it was destroyed. If the building was old and dilapidated by use and decay, its value in that condition should be the ground of recovery. If the policy stipulates that the cash value of the property destroyed or damaged shall not exceed the cost of rebuilding it, and in case of damages from use or otherwise that a suitable deduction shall be made from the cost of repair, the measure of damages is the cost of repairs, if thereby the property is rendered as valuable as before. If less valuable than before, then the difference must be added to the cost; if more valuable, it must be deducted.

22 Under the laws of Massachusetts, no insurance company shall knowingly issue any fire insurance policy upon property within the Commonwealth for an amount which, with any existing insurance thereon, exceeds the fair cash value of the property, nor for a longer term than seven years. If the buildings insured against loss by fire are totally destroyed by fire, the company shall not be liable beyond the actual value of the insured property at the time of the loss or damage.

FAIR CASH VALUE

23 Fair cash value is defined as being the price a willing seller would accept from a willing purchaser, without any undue advantage to either party.

24 The influences which affect the salable values of property are variable and often complicated. It has been said that differences between assessors on questions of valuation of the same class of property are no greater than frequently arise between witnesses in a trial on questions of value. There is no certain, definite standard of values excepting that of money and standard marketable articles. Many influences, tangible and intangible, affect the salable value of property, real and personal, both in city and country, so as to make its real valuation a work of great difficulty, resulting in inevitable inequalities. Mere mistake in judgment is no discrimination.

25 The Massachusetts rule is that the fair cash value is determined upon the basis of a willing seller and a willing purchaser.

26 In New Jersey the rule has been stated as being the ordinary selling and buying price for cash at the time, irrespective of the actual cost of the property, the amount it would realize at a public sale being a good criterion. This rule would seem to

admit of the possibility of some undue advantage to the purchaser, as a public sale might not always be a willing sale on the part of the vendor. On the other hand, the reason for the rule suggesting public sale as a criterion for establishing values is undoubtedly based upon the possibility of inflating values for some reasons, or discounting them for other reasons; also for the purpose of lending aid to a uniform and equal basis of valuation on the ground that the records of such sales of property of the same class, within the territorial limits, will be available to form a standard of comparison.

27 The Massachusetts rule employing the words willing seller and willing purchaser implies the status of equality between the parties which is not in the nature of a forced sale. In a case where a manufacturing property was sold at public auction, on account of the owner's personal wish to sell the property, there is the fact of a willing sale to the purchasers, who paid about one-quarter of the appraised value of the property, as tangible property. If the sale had been made under legal process, by order of the court, it would have been a forced sale, even though the purchase price paid was the same. The price paid under the circumstances may be held as not representing the fair cash value. The sale can be said to more closely resemble a sacrifice by the owner.

COMMERCIAL VALUE

28 The commercial value is the estimated worth of property placed upon it as a business proposition, the chief considerations being the earning capacity of the plant and the amount of profit to be derived from it, both present and prospective. Involved in these considerations are location, facilities for receiving and shipping goods, available supply of help, management and reputation.

29 This value is to be determined by a proper capitalization of the earnings, which for manufacturing properties will fairly represent the market value of the property. It is evident that, as to the owners of a prosperous industry, the commercial value may be above the market value of property of the same class, also that some industry may be below the market value of property of the same class. The method of management may have the weight of influence to determine the commercial value in some cases, or a peculiarly favorable market for the product. These reasons are largely of local value rather than of general value. In the case

of poor management there is a factor that can usually be remedied. In the case of a peculiarly favorable market, the influence of competition, or the changes in fashion, will in time level the particular advantages accruing to the individual plant.

30 Industries may be so located as to involve the expenditure of a larger amount of money in the establishment of their tangible property than other plants more favorably located, owing to some particular advantage gained at the time of their inception. With the advent of new inventions and improved methods, the reason for having the plant in such a locality has passed away. Conditions and environment may result in larger operating expenses reducing the profit. As a business proposition the value of the tangible property can be no greater than that of other plants of the same class more favorably located and costing less; and their commercial value should be determined by an amount equivalent to the charge upon the profits owing to the unfavorable conditions. This is illustrated by many cases where plants have been located in out-of-the-way places to take advantage of some water power. With the advent and increased use of electricity, with the growth of the plant beyond the capacity of the water power, with a growing scarcity of help, and with a more favorable locus for producing the same kind of goods in other places, these plants have been forced to suffer loss and prestige.

31 In valuing industrial property for tax assessment, insurance indemnity or for sale, the commercial value is the leading consideration, as upon it would depend the attitude of the purchaser towards the seller. In determining the value of property upon the basis of net earnings or profits the amount to charge off against the tangible property would be the extra expenses involved in manufacturing the product that could be saved if the plant was working under favorable circumstances. This would not necessarily affect the value of the collateral intangible or incorporeal property, except in so far as the loss of profit might affect the sale of the stock, which might also be affected by loss of profit due to poor management. While poor management may have a very material affect upon the commercial value of an industry, it is subject to correction, and the result of poor management may be felt for some time after the cause has been removed. It is not inherent in the property the same as unfavorable location, the betterment of which would mean the removal of the plant.

32 In valuing the tangible property, its adaptability for other uses is to be considered. This involves the amount of capital which must be expended to put the property into a proper condition to carry on the business for which it may be purchased. In the event of carrying on the same class of business the question of the economical organization of the plant, as compared with other well arranged plants entering into competition would be involved.

33 The uniform and equal valuation of textile, and other industrial property and power plants should by this time be well within the range of possibility, and no one should be any better able to make fair and unprejudiced appraisals than the mechanical or industrial engineer. There is a sufficient amount of information at hand true to the facts of cost, both for installing plants and for operating them, covering all classes, especially of the textile industries. The engineer, who has made certain industries a specialty in all of the processes of mechanical organization, cost of establishing, and cost of operation, should be better qualified than any other man to act as an expert in such valuations. He has a broad field from which to gather his information and a larger opportunity to view the various grades of the same classes of industry. The ethics of his profession should lead him to be an unprejudiced witness before the court. The theory upon which the expert witness is allowed to testify is that he can assist the court to arrive at a just decision between the parties in a controversy, owing to his superior knowledge and experiences. His evidence should be free from the taint of partisanship, and he should stand before the court in the same attitude as an attorney, representing his client, but serving the court. The expert witness is the only one allowed to give opinion evidence, on the theory of his special qualification.

34 No valuation can be made without taking into account the depreciation.

DEPRECIATION

35 Depreciation is the falling off in value. Physical depreciation refers particularly to the lessening value of material things, due to age and wear, and improvements in the mechanical arts. Everything of substance has some intrinsic value, however small.

36 There are several methods of computing depreciation;

some are represented diagrammatically by curved lines, others by straight lines, all of which methods show some variations, but any method adopted may be only approximately correct. Depreciation begins at once, however small, and the actual process will not proceed very far before the work of maintenance begins. Whatever the method or rate of computing the depreciation, if it is sufficient to meet the conditions its relative accuracy is near enough to the truth for all practical purposes. Then it is logical to utilize the simplest form of diagram to represent the loss, which is the straight line computed on the original cost, or the replacement cost if new at the time of making a valuation of property.

37 While it is not the purpose to discuss the several methods of computing the depreciation here, as these discussions may be read elsewhere, the rate of depreciation may be briefly considered.

38 While the rate of depreciation will vary with materials and circumstances, it may be possible to determine a fairly constant rate applicable to property of the same class, used under similar conditions and subject to the same general care and maintenance. In the case of the Chicago Union Traction Company vs. State Board of Equalization, 114 Fed. Rep. 557, it was held that an annual reduction equal to 6 per cent of the current value of cars, tracks and machinery was not excessive as the allowance for current depreciation.

39 In the textile industries, plants may be grouped together in classes, each class being subject to the same economic conditions. In the cotton industry, plants may be classified as duck mills, course sheeting mills based on a standard weight of cloth containing three yards to the pound, medium weight goods mills based on a standard of four yards of cloth to the pound, print goods mills, making goods for the general household trade, and fine goods mills. The mills of each class enter the market to sell their product in competition with the other mills of the same class. Each mill of the same class must strive for the highest efficiency in order to compete successfully with the others, and these conditions have led to a highly scientific development of the industry. In the woolen and worsted trade, the same highly efficient physical organization is being wrought out, until it seems as if there can not be any very radical changes in the methods or general types of machines used in the process of manufacture.

40 In all mechanical and industrial processes the same laws of

growth and development are as potent as are the laws of evolution in the natural growth and development of the physical world. The very nature of the materials we have had to convert into a marketable product for our use has indicated the lines of development for the mechanisms of our industrial processes. Many improvements are yet to be made and perfected.

41 One of the most notable developments and advancements in the art in recent years is the advent and successful operation of the steam turbine, and prior to it the introduction and development of electricity for all purposes. One of the greatest developments yet to come is the method of generating heat without coal.

42 A statement was recently made that no great advancement has been made in the cotton industry for about forty years. This is a cause for surprise and some inquiry. The period from 1870 to 1880 appears to mark the beginning in improved modern buildings, better lighted rooms, better fire resisting construction, more economical arrangements of plants generally. The decade from 1880 to 1890 appears to have been the period of increase of the product of machinery and of a higher efficiency of the plant generally. The decade from 1890 to 1900 appears to have been the time for introducing better organized plants, of giving more attention to the welfare of the employees and tending to raise the standard of quality generally. The last decade has witnessed the introduction and successful operation of electricity generally, better appointed power plants, more efficient machinery and buildings, and the study into the question of more efficient work people.

43 In the decade from 1870 to 1880 the average speed of the front roll of spinning frames on print goods numbers was 64 r.p.m., the average product being about 0.80 lb. in 60 hours. Today the average speed of the front roll for the same numbers of yarn is 122 r.p.m., with the average product of 1.44 lb. per spindle in 60 hours, an average increase of 80 per cent per spindle, with an infinitely better quality of yarn. The product of looms has increased from an average of about 27 lb. per week to an average of 44 lb. per week on print goods. Forty years ago the average product per week gross floor area was about 1 lb. of cloth to 7.25 sq. ft., while today in modern mills the average product is about 1 lb. of cloth per week to 3.5 sq. ft. for a week of 60 hours. These results have been calculated from the records of a number of mills of the earlier period and compared with what is seen in practice today. The increased product of machinery may show

a greater ratio than the increase per square foot of floor area for the reason that in our modern mills more alley room is allowed around the machinery, spinning frames are wider and built to a larger gage than in the earlier days, looms are larger and heavier for the same width of cloth. There is a saving in the card room, as the revolving flat cards used today occupy less room than the double system of flat top cards used years ago, and a large increase in weaving, due to the use of automatic looms.

44 Within the last ten years there have been some marked improvements in the building of machinery. It is built more strongly, on better lines, and is more rugged. This is quite apparent in the case of some steam engines, as compared with earlier types.

45 As to the depreciation rate to charge off against machinery in the case of textile machinery, it is a common thing to find groups of the same kind of machines in a mill varying in age from fifteen to eighteen years: sometimes one sees a group of machines over twenty years old, and it is not uncommon to see single machines from thirty-six to forty years old. As a rule it would be beneficial to the organization to dispense with these old machines. The writer visited a mill sometime ago where there were some old roving frames and was told by the agent that it was more economical to let the old frames remain idle than to try to run them. It is short-sighted policy that keeps such old machines out of the junk heap.

46 By the time spinning frames reach the age of fifteen years they usually begin to require more extensive overhauling than ordinary repairs. The writer has known frames of about this age to require such thorough refitting that the standards were about all that was left of the original machine.

47 During the recent investigation by the tariff board appointed by the President of the United States, some rather interesting data were obtained regarding length of time machinery has been used in worsted and woolen mills, not necessarily indicative of the rate of depreciation of such machinery, although certain inferences may be drawn. Some of these data are given in Tables 1-5.

48 There is nothing in any of the tables given to show the age of any of the individual plants from which these lists were made up, so that no allowance can be made for new industries starting up within the last ten years, but between the ten and twenty year period there is about the same ratio for each period of five years.

and after this period the amount of machinery that has been in operation is relatively small.

49 From personal observation among a large number of textile mills the average rate of depreciation appears to follow along

TABLE 1 YEARS IN OPERATION OF WORSTED MACHINERY, BRADFORD SYSTEM

	Scouring		Carding		Combing		Gilling		Drawing	
	No.	Per Cent	No.	Per Cent	No.	Per Cent	No.	Per Cent	No.	Per Cent
Less than 5 years	11	16.4	173	26.3	53	14.0	137	18.5	797	32.9
5 years but less than 10 years	26	38.8	143	21.7	136	35.8	214	29.0	628	26.0
10 years but less than 15 years	17	25.3	76	11.5	18	4.7	68	9.2	126	5.2
15 years but less than 20 years	5	7.5	61	9.3	46	12.1	66	8.9	434	17.9
20 years but less than 25 years	5	7.5	155	23.6	43	11.3	116	15.7	92	3.8
25 years but less than 30 years	2	3.0	31	4.7	47	12.4	71	9.6	214	8.9
30 years and over	1	1.5	19	2.9	37	9.7	67	9.1	129	5.3
Total machines	67	100	658	100	380	100	757	100	12,500	100

TABLE 2 YEARS IN OPERATION OF WORSTED MACHINERY, SPINNING AND WEAVING

	Mules		Frames		Looms*	
	No.	Per Cent	No.	Per Cent	No.	Per Cent
Less than 5 years	92	24.9	273	20.3	4125	33.4
5 years but less than 10 years	121	32.7	559	41.5	1244	10.0
10 years but less than 15 years	12	3.2	109	8.1	2009	16.3
15 years but less than 20 years	34	9.2	76	5.6	2072	16.8
20 years but less than 25 years	100	27.0	98	7.3	2053	16.7
25 years but less than 30 years	11	3.0	116	8.6	396	3.2
30 years and over	0	0	115	8.6	438	3.6
Total machines	370	100	1346	100	12,337	100

* Includes all looms for both worsted and woolen mills.

these general lines. Up to about fourteen years of use the machinery contains practically the original parts with which it was furnished, subject to the ordinary repairs that an economical mill will keep up. Of course accidents or some unlooked-for event

TABLE 3 YEARS IN OPERATION OF WOOLEN MACHINERY

	Cards		Mules	
	No.	Per Cent	No.	Per Cent
Less than 5 years	40	9.6	47	8.0
5 years but less than 10 years	62	14.8	204	34.7
10 years but less than 15 years	68	16.2	143	24.3
15 years but less than 20 years	16	3.8	72	12.3
20 years but less than 25 years	36	8.6	47	8.0
25 years but less than 30 years	32	7.6	53	9.0
30 years and over	165	39.4	22	3.7
Total machines	433	100	588	100

TABLE 4 RELATION BETWEEN DOMESTIC AND FOREIGN MADE MACHINERY, GIVING THE NUMBER OF MACHINES IN PLANTS LISTED

Country	Scouring	Cards	Combs	Gills	Drawing	Spinning	Looms
United States	37	730	57	37	273	617	9517
England	18	271	323	210	2488	1486	2713
Germany	7	90	225	90	192	195	107
France	5	0	22	32	36	6	....
Total	67	1091	627	369	2989	2204	11337

TABLE 5 AVERAGE YEARS IN OPERATION OF WORSTED MACHINERY LISTED IN TABLES 1 AND 2

Years in Operation	Average Per Cent of Total Machinery
Less than 5 years	26
5 years but less than 10 years	29
10 years but less than 15 years	10
15 years but less than 20 years	11
20 years but less than 25 years	13
25 years but less than 30 years	6
30 years and over	5

may happen to a machine to cause some extraordinary expense. From the tenth to the fifteenth year there is likely to be some very general overhauling of the machinery, renewal of vital parts, and perhaps the substitution of one or more machines in a group of similar machines. The whole organization is toned up generally. Conditions will run along for the next ten years or so under the ordinary mill care when there begins to be a very general reorganization of the plant.

50 The inference might be drawn from Tables 1-5 that the general, or average, depreciation may follow the results shown, and this does not seem an unreasonable, although it may not be the true conclusion. Assuming that the inference is approximately true, it seems that a depreciation annuity of 6 per cent on the cost of the machinery is not too much to set aside in order to provide a proper and conservative amount with which to replace textile machinery. This would allow for a partial replacement at the end of the 15 year period, and a general renewal of the plant at the end of 25 years. If the plant is in a good condition at the end of 25 years and can continue to operate for a few years longer, the gain is so much greater.

51 Industrial buildings are planned for certain specific purposes. The organization of the plant is first worked out, the departments arranged most economically, then the walls and roof put over the whole.

52 Under proper conditions, buildings will last for hundreds of years, but with factory buildings, the conditions are such that it is not expected to have a building last much beyond a certain limited time, owing to the change in methods. It is noticeable, however, that the buildings for manufacturing purposes of recent years are more substantially built than they were 50 years ago.

53 In looking over the records of existing textile industries, buildings in use which are over 50 years old are not very numerous; there is a larger number of those 40 years old, and a rapidly increasing number in comparison as the age diminishes. The old buildings are wholly unsuited for modern practice. In fact, many buildings built 25 years ago are not suitable for a proper arrangement of modern machinery.

54 With more rugged and better built machinery and stronger and better made buildings it is possible that the depreciation will not be so rapid as it has been in the past. To a very large extent

depreciation is due more to rapid advance in the mechanical arts than to age and wear. The proper maintenance of a plant can retard the depreciation due to age and wear very materially. Depreciation due to advance in the arts is, in the author's opinion, one of the essential factors to be considered in the use of reinforced concrete buildings, as this type of building is not so readily adapted to be modified or changed over to meet new conditions as is the regular mill construction type. It may be, however, that radical changes are not so likely to come in the future as they have been in the past, but this seems a short-sighted view.

55 In depreciating factory buildings, a rate of about 2 per cent a year does not seem excessive, compared with the useful life of the older type of building, while about 1 per cent a year may be large enough to allow for the depreciation of the modern buildings.

56 In making up valuations of textile plants the physical depreciation for the preparatory, spinning and weaving machinery should be taken at about 4 per cent a year, while for dye-house and wet-finishing machinery the wooden parts of machinery have to be entirely renewed on an average of every five years, so that the depreciation on these machines after three years is very nearly a constant rate for a going concern, and can be taken at about 25 per cent of the replacement cost.

CAPITALIZATION

57 To capitalize is to convert an income, annuity, etc., into a single payment or an equivalent capital sum.

58 Under the subject of capitalization we must take into account what is included in the fixed charges on the plant. It is a very easy thing in handling capitalized values not to include some element in the calculations that belongs there and so raise false values, or vice versa.

59 The fixed charges usually include taxes, insurance and interest on current liabilities. The factors subject to capitalization are the annuities for depreciation, the annuities to discharge the funded debt and a fair profit to be derived from the enterprise.

60 The rate of capitalization is not always easily determined and may vary under varying circumstances for the same class of property.

61 In making up the commercial valuation of the railway

operating property of the United States in 1904 the rates of capitalization vary from about 3½ per cent to about 10 per cent based on the net earnings of the railroads, and the total of these capital amounts represented the wealth of the railroad property in this country at that time. The whole subject is discussed in Bulletin 21, Bureau of the Census.

62 In determining the rate of capitalization of values for industrial properties, Table 6 shows the factors which may be taken. The table is based on the assumption that the economic life of buildings is 50 years, and that the annuities are invested to earn a return of 4 per cent compound interest, the buildings

TABLE 6 FACTORS IN DETERMINING RATE OF CAPITALIZATION

	Annuities	Percentage of Contributions		
1	Buildings depreciated at 2% a year	0.25	0.25	0.25
2	Machinery depreciated at 4% a year	1.50	....	....
3	Machinery depreciated at 5% a year	....	2.10	....
4	Machinery depreciated at 6% a year	....	....	2.65
5	Annuity to discharge funded debt, say	1.12	1.12	1.12
6	Fair net profit	10.00	10.00	10.00
	Total	12.87	13.47	14.02

being about three-eighths and the machinery about five-eighths of the cost of the tangible property.

63 As the plant grows old and improvements are made in methods and efficiency the net profit will decrease. Assuming this decrease to follow at about the same rate as the average physical depreciation of the property, there will come a time when it will cease, unless proper precaution is taken to provide a fund large enough to maintain the property by renewals from time to time; and the time will always come when some general reorganizing and renewals are necessary. It seems justifiable to take the depreciation fund at 6 per cent per year of the cost for textile machinery, and following similar periods of usefulness to allow for not only the renewals which must be provided for in due course, but for renewals due to improved and more economical machinery.

64 A gross income of 20 per cent should be conservatively assured from a new industry not only to cover the annuity

charges and the profit expected to be realized from year to year, but also the extra expenses of establishing the plant and the risk involved in floating the enterprise.

65 In capitalizing values the rate of 10 per cent is often used. If this includes the annuity charges for depreciation, then the allowance for the net profit is only about 5 per cent. If it is assumed to include the net profit only, it is a more conservative assumption.

66 The method of capitalizing a possible saving to an industry to determine comparative values is often not correct, as the capitalized values lead to exaggerated conclusions which are wrong. What percentage of saving could be effected by comparing the costs of installation and operation between two processes should be ascertained, and the values then determined from the cost of installation. If the saving amounted to 15 per cent, the cost of making a change would be considered justifiable; if to 10 per cent, the cost of making the change would not be considered justifiable, except as modified by the question of necessity, as by the renewal of an old plant under similar conditions. A saving of 10 per cent might warrant a change in methods. Thus in ascertaining comparative values by the method of capitalizing possible savings, it would seem as logical to use a rate of 15 per cent on the basis of the above reasoning.

67 A low rate of capitalization may lead to exaggerated and wrong conclusions. It is easy to mistake the principle involved, and rate the value of tangible property on the negative side of the equation. A small variation in the rate results in large variations in the values. The values for tangible property should always appear on the positive side, as all material property has some intrinsic value, however small.

68 As an example of this principle we will assume a cotton mill equipped with 500 automatic looms and 500 plain looms. The plant is making a class of goods where automatic looms can be used to advantage. The product of the automatic looms is about 45 lb. of cloth per week per loom, that of the plain looms about 40 lb. of cloth per week per loom, or say about 1,000,000 lb. per year for the 500 plain looms.

69 By using 444 automatic looms in place of the 500 plain looms a net saving of about 1¼ cents per lb. of product can be made, or a total yearly saving of $1{,}000{,}000 \times 0.0125 = \$12{,}500$. The cost of 444 automatic looms at $168 each, including installa-

tion, would be $74,592, and if we allow a salvage value of $10 per loom for the plain looms the net cost of installing the automatic looms would be $69,592. A yearly saving of $12,500 would be about 18 per cent of the cost of installing automatic looms, so that the mill would be warranted in making the change.

70 The replacement cost of 500 plain looms, if new, would be about $30,000. If we capitalize the saving of $12,500 in favor of automatic looms at the rate of 15 per cent, the capital value equals $83,333. We have already allowed a salvage value of $5000 for the plain looms so it is evident that the only value of the capitalized saving in favor of automatic looms to be applied to the plain looms is an amount having the positive value of $5000.

71 Again, assume that the cost of installing the automatic looms is $74,592. The replacement cost of the plain looms if new is about $30,000, which depreciated for age and wear is at present $5000, with no rebate by the loom company as an inducement for taking these looms out and replacing them with automatic looms. The gross cost chargeable to installing automatic looms would be about $79,592, including the value of the plain loom, and a yearly saving of $12,500 would be about 16 per cent of this cost in favor of automatic looms, which indicates that the investment would be warranted.

72 With reference to the commercial value of the plain looms, the capital value of the saving that could be made by using automatic looms can not be applied to reduce the value of the plain looms by an amount equal to more than their depreciated replacement cost, as a positive value, or $5000.

73 One of the hardest experiences is to recuperate an old plant after it has been allowed to reach such a state of physical depreciation as to become profitless, and there has been no depreciation fund provided to take care of future contingencies. Such conditions are probably more likely to occur in privately owned industries than in corporations. There are numbers of cases where plants in such a condition have been taken over and operated, but this means the sinking of new capital.

74 A case coming under the author's observation is that of an old mill which had reached the limit of its usefulness as a producer. Under a proper management, the old buildings were replaced by a substantial modern mill. During a period of depression the conclusions as to the wisdom of building a new mill

were that with the old buildings the plant would have been forced to close, owing to the stress of circumstances. The light was so much better in the new mill that the goods formerly passing as first quality in the old mill were not equal to the second quality in the new mill. Owing to trade conditions, it had been necessary to cut wages of the operatives twice, yet with the reduced wages the best operatives were able to earn more money in the new mill than they had been able to earn in the old mill before any reduction in wages had been made.

METHOD OF VALUING A PLANT

75 In the event of making a valuation of property, the first essential thing is for the appraiser to go over the property as a whole, to see how the plant is arranged, and how the arrangement compares with other plants making the same general kind of manufactured product. This survey will usually disclose any special features that may be unfavorable to a more economical arrangement, or where certain alterations might be made with benefit to the plant, as the organization and arrangement should be considered one of the vital features in ascertaining values.

76 It is not enough to say that the physical property of an industry can be replaced for a certain sum of money, or that it costs a certain amount and is to be valued at a certain other sum of money after allowing for a certain depreciation due to age, wear, obsolescence, etc. An investigation must be made into the working of the plant, comparing it with other plants in competition with it. Its location must also be considered.

77 Any extra cost of operation that can be eliminated by a proper reorganization must be accounted for and set off against the value. Sometimes the cost of reorganizing a portion of the plant is greater than the expense of making the change would warrant, but the poor feature will remain nevertheless a detriment to the plant and affect the value.

78 Inventories of property can be given by the owners of the plant, but these inventories should always be checked up by a person familiar with the business, who knows the process of manufacturing the goods produced, the machine used in each process, and how they should be arranged in each department, and who is capable of taking a birdseye view of the whole property and seeing it as a working whole, as well as segregating the several parts.

79 For this reason the engineer who is familiar with certain lines of industry is better able in general to take an unprejudiced as well as a more comprehensive view of the subject, owing to his wider experience and more special knowledge of engineering than a person engaged in commercial lines.

INCREASE OF BORE OF HIGH-SPEED WHEELS BY CENTRIFUGAL STRESSES

WITH FORMULAE FOR FORCE AND SHRINK FITS

By Sanford A. Moss

ABSTRACT OF PAPER

Unless proper precautions are taken, a high-speed turbine wheel expands at the hub a sufficient amount to cause an appreciable increase in the bore so as to make the wheel slightly loose upon the shaft, soon causing distress.

There is given in the paper a general discussion of stresses in a rotating wheel; a mathematical method for computation of the increase in bore of a rotating wheel due to centrifugal stresses; an investigation of the actual numerical value of the increase of bore under various practical conditions; an experiment giving an actual measurement of the increase in bore of a wheel at high speeds, showing close agreement with computed values; a description of methods of preventing damage due to the increase of bore by using large amounts of force fit; and a mathematical treatment of force fits.

The increase of circumference of the bore is proportional to the original circumference and the tangential stress and inversely as the modulus of elasticity. In the experiments the readings showed that the bore expanded at each speed an amount almost exactly equal to the computed values.

For a wheel with 12 in. bore and 30,000 lb. maximum stress, which is permissible with special grades of steel, this increase of bore comes out over 0.01 in., which gives an appreciable loosening of the shaft fit.

A complete treatment of the subject of force and shrink fits based on the maximum shear law is given for the first time so far as known.

When the hub is in place on the shaft, it is expanded by a certain amount and the shaft is also compressed by a considerable amount. The ratio of expansion of the hub to the total force fit is the same for a given wheel and shaft regardless of the amount of the force fit. A curve is given showing values for various cases. Curves are also given showing values of radial stress at the bore of the hub which is also the pressure between hub and shaft, and the maximum effective stress at the bore of the hub which is usually the greatest stress in the system. The usual case of a shaft with a comparatively small hole is practically equivalent to a solid shaft for these matters.

For an average case the hub bore expands about 0.6 of the amount of the force fit and for a force fit of one mill per inch of bore the radial stress is about

13,000 lb. per sq. in. and the effective tangential stress is about 27,000 lb. per sq. in. For other amounts of force fit the stresses are in proportion.

For the average case above mentioned the pressure to force the hub on the shaft is about 1560 lb. per in. of bore and per inch of hub length for a force fit of one mill per inch of bore.

INCREASE OF BORE OF HIGH-SPEED WHEELS BY CENTRIFUGAL STRESSES

WITH FORMULAE FOR FORCE AND SHRINK FITS

By Sanford A. Moss, West Lynn, Mass.

Member of the Society

Recent advances in high-speed steam turbines and the apparatus driven by them have opened up a great many new problems. One of these is the fact that, under certain circumstances, unless proper precautions are taken, a high-speed turbine wheel, or the like, expands at the hub a sufficient amount to cause an appreciable increase in the bore, so as to make the wheel slightly loose upon the shaft and capable of pounding back and forth. This pounding causes a mechanical increase of the looseness, so that the machine soon gives evidence of distress.

2 So far as the writer knows, the first one to direct attention to this problem, to suggest means of investigating it, and to offer a solution, was Richard H. Rice, chief engineer of the Lynn works' turbine department, General Electric Company. The present paper gives an account of work on the problem done under the general direction of Mr. Rice by various members of his department.

COMPUTATION OF INCREASE OF BORE OF A ROTATING WHEEL

NATURE OF STRESSES IN A WHEEL OR RING

3 It is the purpose first to give a general account of methods of computing stresses in circular bodies. When there is rotation this becomes a very intricate subject and details and formulae must be sought in works specially devoted to it.

4 It is assumed that the disk is thin, that is, that the thickness is negligible compared with its diameter. Then conditions are the same at all points on a line parallel to the axis. This is not

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strictly true, as the central part of a hub is strained differently than the ends. However, in most cases this effect is slight.

5 At each radius in the disk there are two stresses, one in the radial direction denoted by S_r and one in the tangential direction denoted by S_t. If we consider the effect of these stresses on the metal of a thin ring at the interior of the disk, we find the circumference of the ring to be increased mainly by virtue of the tangential stress, and the thickness mainly by the radial stress. However, when a metal is extended in one direction by a stress there is a shrinkage in a direction at right angles, the ratio of the deformations being called "Poisson's ratio," which we will denote by V. For ordinary kinds of steel this has a value of 0.3. That is, the increase of circumference of a thin ring in the interior of the disk is less than that which would be produced by the tangential stress if acting alone, by an amount which is 0.3 of the extension which would be produced by the radial stress acting alone. Similarly the increase in the thickness of the ring is less than that produced by the radial stress acting alone. In any case, the ratio of stress to extension per unit length produced by it if acting alone, is given by the modulus of elasticity E, usually taken as 29,000,000 lb. per sq. in.; or, if l is the total extension in a distance L, we have the well-known law

$$E = \frac{S}{\frac{l}{L}} \quad [1]$$

The combination of the above principles with the laws of the centrifugal forces on the various elements of the disk gives formulae which enable computations to be made of the stresses in the disk at all points during rotation.

6 In order to determine the safety of a structure where there are stresses in more than one direction, it is necessary to find an effective stress which corresponds so far as yielding is concerned to the stresses in a specimen of the same material subject to simple tension in a testing machine. It is now almost universally agreed that Guest's maximum shear law is the proper one. This states that in cases such as the present, where there are stresses in two directions only with zero stress in the third, and where the stresses are either both tension or both compression, the greater of the two stresses is the effective stress. This effective stress is equal to a simple stress which would produce like results so far as yielding, safety, elastic limit, etc., are concerned.

7 This is not true, however, when there are stresses in all three di-

rections, as in the case with a rotating disk of appreciable thickness where there is change of stress in the axial direction; or with two stresses with opposite sign, as in the stresses in the hub due to force fits, discussed in Pars. 29-39. In such cases the maximum algebraic difference between the greatest and least of the stresses existing at a point taken with proper sign gives the effective stress or equivalent simple tension.

8 In most cases of properly designed rotating wheels with a hole at the center the maximum stress is the tangential stress at the bore. This is, therefore, the stress which is to be considered in order to ascertain if the wheel is safe and is further the stress which must be determined in order to discuss the problem of the present paper.

FORMULA FOR INCREASE OF BORE

9 As we proceed outward from the bore in a rotating wheel, the radial stress mounts up rapidly, due to the fact that the centrifugal forces of the outer portions of the disk pull outward on the inner portions. The nearer the center the less the material in the inner circle, and therefore the less resistance to radial elongation. At the bore there is no force whatever resisting radial elongation, so that the radial stress becomes zero as this inner diameter is reached. We have then on the inner wall of the cylinder forming the bore, material which is stressed only in the tangential direction and the increase of circumference is exactly the same as if we had a straight bar of the same length stressed an amount equal to the tangential stress at the bore. This is expressed by the simple relation which is the fundamental equation of our problem:

$$X = \frac{S_{to}\, D_o}{E} \qquad [2]$$

where

X = increase in diameter in inches of the bore, due to the centrifugal stresses at a given speed

S_{to} = tangential stresses at the bore in pounds per square inch at the same speed

D_o = original diameter of bore in inches

E = modulus of elasticity, 29,000,000 for steel

10 It can be shown that the stresses at any point in a given wheel or at a similarly situated point in any other wheel of similar proportions, are directly proportional to the square of the peripheral wheel speed. Hence the increase of bore in which we are interested varies directly with the square of the peripheral wheel speed with

a given wheel. In any case of the same or geometrically similar wheels, the increase per foot of bore increases directly with the square of the wheel speed.

NUMERICAL MAGNITUDE OF INCREASE OF BORE

11 The allowable values of the tangential stress at the hub are quite high, since a centrifugal load is one of the easiest loads which can be applied and is of the nature of a pure dead load in a beam. It is not possible to apply or remove a centrifugal load suddenly,

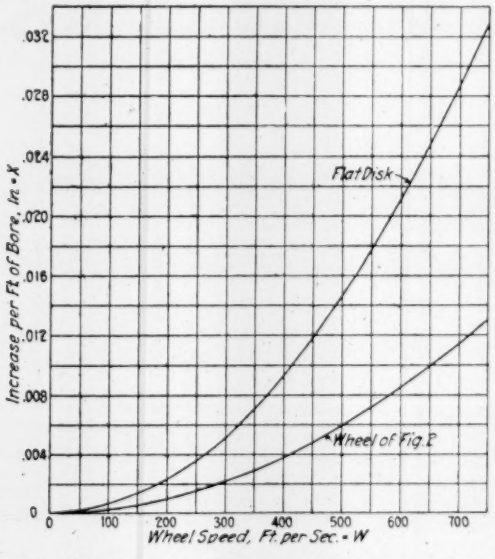

FIG. 1 INCREASE OF BORE OF A ROTATING WHEEL

Ratio of Diameter of Bore to Diameter of Disk = 0.353. This Ratio may be changed considerably without appreciably affecting the Results.

Approximate Dead Weight per Pound of Disk (equal in Both Cases) = 100,000 Lb. per Ft. of Disk Circumference at 534 Ft. per Sec.

but only in the most gentle manner. Also the maximum stress usually occurs only in a single set of fibers which are reinforced by adjacent fibers with decreasing stresses. There need be no "factor of ignorance" and maximum stresses can be known with as great accuracy as it is desired to give to the mathematical computations. In many cases 20,000 lb. per sq. in. is a conservative stress. In some cases with special grades of steel 30,000 lb. per sq. in. is possible. For a wheel with 12 in. bore and 30,000 lb. maximum stress the increase in bore computed from [2] is 0.0124 in., which is an appreciable

amount. For other stresses and other dimensions of bore the increase of bore is in direct proportion. When the stress is 29,000 lb. per sq. in. the increase in bore is 1 mil per in. of bore.

12 Fig. 1 gives numerical values of increase in bore for various wheel speeds for two different shapes of disk. The upper curve is for a disk of uniform thickness. This shape is only possible with comparatively low wheel speeds and the stresses become excessive with speeds ordinarily used in turbine practice. The lower curve is for a disk of the more efficient shape shown in Fig. 2. In both cases the computations are made for a turbine bucket wheel with a load

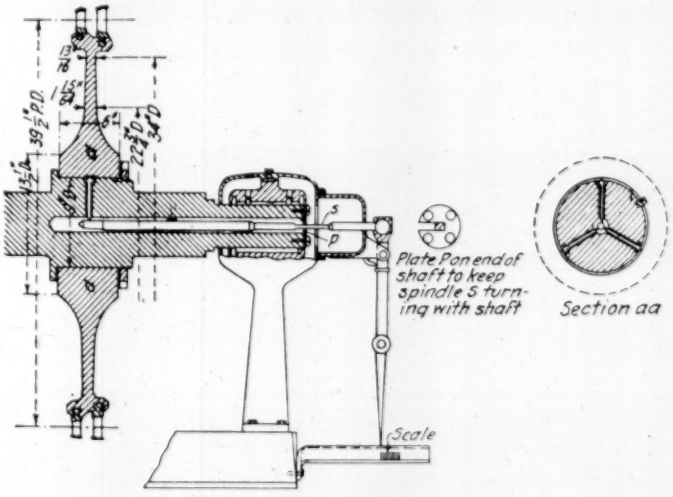

FIG. 2 DIAGRAM OF DEVICE FOR MEASURING THE INCREASE OF BORE DURING ROTATION

due to centrifugal force of buckets at 3600 r.p.m. of 100,000 lb. per ft. of circumference. The shapes of the curves are affected somewhat by the magnitude of this outer dead load. In both cases the ratio of bore to disk diameter is 0.353. The curves are not greatly affected by difference in this ratio. The curves give very nearly the values of increase in bore per foot of bore for various wheel speeds for a disk of uniform thickness, which is about as poor a shape as would ever be used, and for a disk of what is probably as efficient a shape as would ever be used, with an average bucket load. The increase in bore for any shaped disk whatever will probably lie between the two curves.

13 The curves of Fig. 1 were constructed as follows: The wheel

whose dimensions are given in Fig. 2 is the wheel of a standard steam turbine with a web diameter of 34 in. In the course of the design of this wheel the maximum stress was computed for the rated speed, 3600 r.p.m., and came out about 20,400 lb. per sq. in. From the value of this stress and formula [2], the increase in bore at the rated speed was found. The effective bore, taking account of keyways, was 12 in. From the law that the stress and hence the increase in bore varies directly with the square of the wheel speed, the lower curve of Fig. 1 was drawn, taking bore as 12 in., however.

14 Next was taken the case of the upper curve for a flat disk. From the formulae for stresses in flat disks given in Stodola's "Die Dampfturbinen," fourth edition, pp. 248 and 249, can be deduced the following formula for the tangential stress at the bore of a flat disk S_{to}, where S_{r1} is the radial stress at the circumference due to bucket load, W is the wheel speed in feet per second and R is the ratio of bore diameter to web diameter.

$$S_{to}=0.0261W^2\ (0.7R^2+3.3)+\frac{2S_{r1}}{1-R^2} \quad \ldots\ldots\ldots\ldots [3]$$

15 This is for a steel disk whose modulus of elasticity is 29,000,000 lb. per sq. in., Poisson's ratio 0.3 and density 0.28 lb. per cu. in.

16 The centrifugal force due to the bucket load on the previous disk, 100,000 lb. per ft. of circumference, was taken as being applied to a flat disk of about the same weight, giving $1\frac{1}{4}$ in. axial width, and 34 in. web diameter. This gives a radial stress S_{r1} of 6670 lb. per sq. in. at the circumference of the flat disk. The ratio of the diameter of bore to the web diameter R was taken as in the previous disk 0.353.

EXPERIMENTAL MEASUREMENT OF INCREASE OF BORE OF A ROTATING WHEEL

17 The magnitudes of the increase of bore in a number of actual turbine wheels, as determined by substituting computed stresses in equation [2] came out so large that it was decided to make an actual measurement of the increase in bore when a wheel was rotating.

18 Fig. 2 shows the apparatus diagrammatically and Fig. 3 the results obtained. The turbine wheel was mounted on a shaft with a light force fit. A delicate micrometer arrangement was provided so that change in bore of the wheel while it was rotating with the shaft could be observed. The wheel was gradually increased in speed and meanwhile the readings of the micrometer taken. These

showed that the bore expanded at each speed an amount almost exactly the value computed by the methods previously given. The micrometer arrangement proved very positive and there was no uncertainty regarding the measurement of the increase of bore. The maximum amount measured was about 0.007 in. The observations were accurate to about 0.0005 in.

19 The wheel was forced on the shaft with a force fit of 0.0014 so that it was loose on the shaft by about 0.0056 at maximum speed. The duration of the tests was not great enough to show any difficulty due to this.

MEASURING DEVICE

20 The measuring device shown in Fig. 2 consisted of three pistons sliding in radial holes in the shaft pushed outward by means of a tapered plug moving axially in a hole in the center of the shaft. The motion of the central plug was given by means of a lever with a ball and socket joint fulcrumed so as to give considerable multiplication and read by a pointer passing over a scale. A displacement of the pointer by 1 in. corresponded to an increase of bore of 0.004 in. The position of the pointer was certain to within about $\frac{1}{8}$ in. motion along its scale, corresponding to 0.0005 in. change of bore. While the speed was being varied an attendant kept his hand on the pointer, moving it back and forth a slight amount so as always to keep it just in the proper position. As the speed varied from low values to maximum value the pointer moved a distance of about 2 in. so that the change in bore was quite perceptible.

21 Owing to the fact that the increase of bore varies with the square of the speed there was no perceptible change below about 750 r.p.m., and this was the lowest speed used. The highest speed was 3900 r.p.m.

CALIBRATING SLEEVE TEST

22 In order to find the relation between the readings of the pointer and the actual values of increase of bore, and also to demonstrate that the apparatus would give correct measurements at high speed, a calibrating sleeve was made. This was a thin sleeve bored out at each end so as to be a sliding fit on the shaft and with two different bore diameters in the central portion, either of which could be brought over the pistons of the measuring device. Readings of the pointer were taken at each of these bores, stationary and rotating, at various speeds. The stationary readings gave the motion of the pointer for the known change of bore calibrating the

entire apparatus. The readings while rotating showed that the instrument gave correct readings at all speeds.

TEST WITH ROTATING WHEEL

23 Next the rotating wheel was pressed on the shaft. A number of readings of the diameter of shaft and bore of wheel showed that the average amount of force fit was 0.0017 in. After this, a pair of outside calipers was set to fit the shaft at various points and a pair of inside ones set to fit the wheel bore. The difference

TABLE 1 READINGS OF POINTER IN TESTS WITH ROTATING WHEEL, CONDUCTED MARCH 11, 1911

	Test No. 9		Test No. 10		Test No. 11		Test No. 12		Test No. 13	
R.P.M.	Up	Down	Up	Down	Up	Down	Up	Down	Up	Down
0	2.61	2.62	2.62	2.63	2.62	2.63	2.63	2.62	2.61	2.62
750	2.61	2.54	2.63	2.59	2.63	2.59	2.64	2.62	....	....
1050	2.64	2.58	2.70	2.68	2.68	2.67	2.66	2.70	....	....
1200	2.69	2.63	2.74	2.72	2.72	2.74	2.72	2.75	....	....
1500	2.72	2.70	2.80	2.82	2.82	2.80	2.84	2.90	....	....
1800	2.78	2.79	2.90	2.96	2.92	2.90	2.96	3.00	....	....
2100	2.84	2.89	3.03	3.10	3.02	3.10	3.11	3.20	....	....
2400	2.99	3.14	3.18	3.25	3.18	3.24	3.26	3.40	....	....
2700	3.16	3.30	3.32	3.44	3.32	3.48	3.48	3.56	....	....
3000	3.30	3.45	3.54	3.63	3.54	3.63	3.70	3.75	....	....
3150	3.50	3.64	3.70	3.78	3.75	3.85	3.85	3.85	....	....
3300	3.65	3.70	3.85	3.94	3.85	3.98	4.00	4.00	....	....
3450	3.80	3.80	4.01	4.08	4.00	4.08	4.10	4.08	....	....
3600	4.00	4.00	4.13	4.20	4.15	4.18	4.24	4.20	....	....
3750	4.12	4.12	4.24	4.28	4.30	4.29	4.34	4.33	....	....
3900	4.30	4.30	4.34	4.34	4.38	4.38	4.38	4.38	4.38	4.38

between the two calipers was then found by means of a thickness gage. The mean of these observations gives 0.0011 in. as the amount of force fit. The mean of these two values 0.0014 in. is considered to be the actual amount of force fit. It was necessary to know the amount of this force fit in order to make comparisons between the computed and observed amounts of increase of bore of the rotating wheel, since the wheel at zero speed is expanded somewhat.

24 Fig. 2 is a diagram of the arrangement of the apparatus when the wheel was on the shaft. Readings of the pointer were taken at various speeds as shown by Table 1 and the results plotted in Fig. 3. The pointer readings, as well as the corresponding values of actual increase in bore as shown by the previous calibrations, are given

on the left. Owing to the fact that the wheel was forced on the shaft, the measured diameter of the wheel at zero speed is greater than the actual free diameter of the wheel. By methods given in Pars. 29-39, it was computed that the wheel was expanded by the force fit one-half of the difference in free diameters, the remainder being compression of the shaft. Hence the base line for increase in bore of the wheel from the free condition is the lower line of Fig. 3, which is 0.0007 in. below the base line for increase in bore from the

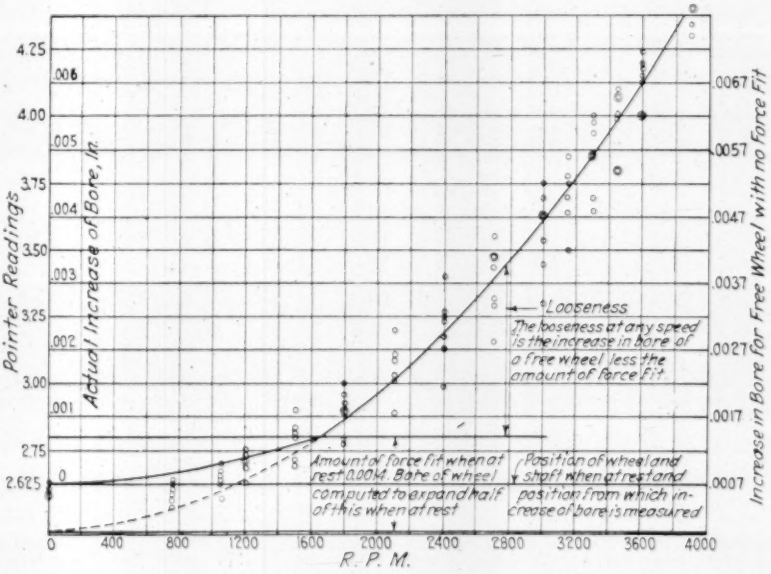

FIG. 3 INCREASE IN BORE OF A ROTATING WHEEL

Points from Test with Rotating Wheel (Table 1); Curve drawn from Computations.

actual initial value when on the shaft. The computed curve for the increase in bore for the free wheel at the various speeds is drawn in Fig. 3 beginning with the lower base line. At that speed where the expansion of the free wheel is equal to the amount of the force fit, 0.0014 in., the shaft is just released from its compression and wheel and shaft are just of the same diameter. For speeds beyond this point the wheel expands exactly as if there was no initial force fit. The wheel is loose on the shaft by whatever increase in bore occurs beyond this point. The observed points are seen to fit very closely the computed curve. For all speeds below that at which the wheel and shaft just touch there is compression of the shaft and expansion

of the wheel by the force fit, the amount being successively less as the speed increases. The effective force fit at any speed is the difference between the force fit at rest, and the expansion of the free wheel at the given speed. The effective force fit is, therefore, the difference between the dotted curve and the upper horizontal line. As shown later, when there are different amounts of force fit with a given wheel and shaft, the expansion of the wheel is always a constant fraction of the amount of force fit. This is one-half in the present case. Hence the curve showing the increase in bore of the wheel,

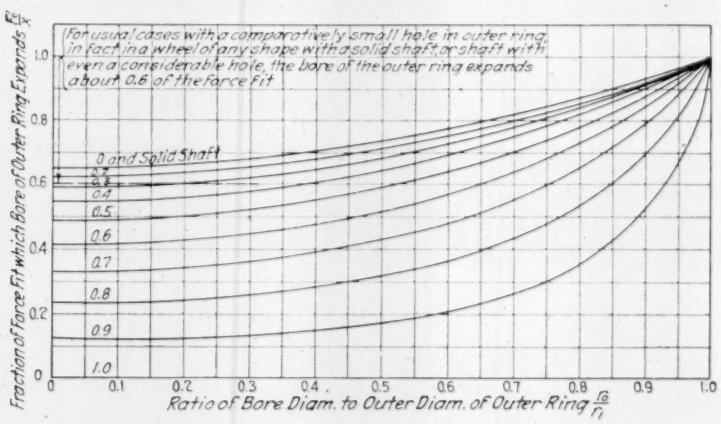
(For usual cases with a comparatively small hole in outer ring, in fact in a wheel of any shape with a solid shaft or shaft with even a considerable hole, the bore of the outer ring expands about 0.6 of the force fit
0 and Solid Shaft
0.4
0.5
0.6
0.7
0.8
0.9
1.0
Fraction of Force Fit which Bore of Outer Ring Expands $\frac{EF}{X}$
Ratio of Bore Diam. to Outer Diam. of Outer Ring $\frac{r_2}{r_1}$

FIG. 4 RATIO OF EXPANSION OF BORE TO FORCE FIT

Figures on Curves denote Ratio of Diameter of Hole in Inner Ring to Diameter of Bore of Outer Ring $\frac{r_2}{r_0}$

as the force fit is gradually relieved, is drawn halfway between the dotted curve and the upper horizontal line.

25 In the present case the amount of the force fit is comparatively small. It was originally intended to repeat the experiments with successively increasing amounts of force fit. The measuring apparatus was arranged so that it could be torn down and reassembled without change in the absolute initial value of the readings. This was done with the idea of actually measuring the change in the initial bore of the wheel when put on the shaft with different amounts of force fit, giving curves in Fig. 3 which would start with successively greater amounts of force fit and meet the curve of a free wheel higher up. However, the results of the first experiment agreed so closely with the theory and everything was

so obvious after study of the subject that no further experiments were made.

DESIGN OF WHEELS SUBJECT TO INCREASE OF BORE BY CENTRIFUGAL STRESSES

26 The method used to prevent difficulty due to loose wheels is to force the wheel on the shaft with a force fit, or difference in diameters between wheel bore and shaft, greater than the computed amount of increase of bore at maximum speed. In some cases this has resulted in large amounts of force fit requiring very heavy pressures for forcing. Special material is necessary for the bushings between the wheel and the shaft. However, all of these points have been successfully met and ill effects due to loosening of the wheels wholly eliminated.

27 In some cases the wheel hubs do not fit along their entire length but only in places. The radial pressure when the fit only occurs for a short distance is obviously increased by the ratio of the total length of hub to the actual length of the fit. The numerical values of this radial pressure on the bushings, computed by methods discussed in Par. 33, come to very high figures in many cases. Of course, some relief is offered by the fact that this pressure is pure compression and that the metal of the bushings is constrained on nearly all sides so that the resistance to flow is considerable. Under such circumstances much higher values of compression stress can be used than when there is no resistance to the deformation which the compression tends to cause. There is reason to believe that in the ideal case of a piece subject to pure compression in every direction, there would be no failure even with an infinitely large stress.

28 It is to be remarked that when the force fit is just sufficient to prevent loosening at full speed and the shaft is solid, the effective stress in the hub due to the force fit when at rest, is exactly equal to the stress at full speed.

COMPUTATION OF FORCE AND SHRINK FITS WITH STEEL

29 A complete treatment of the subject based on the maximum shear law previously mentioned is given for the first time so far as known. The general nature of ring stresses is in Pars. 3-8. The mathematical derivation of the formulae is given in the Appendix.

30 It turns out that a wheel hub is affected only for a very short distance outward so that the nature of the outer portion has little or no influence. The condition near the hub in a flat disk with an

outer diameter only ten times the diameter of the bore is practically the same as in a disk with an infinitely large outer diameter. We therefore consider only the case of a ring forced upon another ring of equal axial length. We estimate the outer diameter of the outer ring so as to make it equivalent so far as expansion is concerned to a given wheel of any shape. For instance, for a solid wheel comparatively thin at the outer portion, we would estimate the equivalent flat disk to have one-half or one-third the diameter of the wheel.

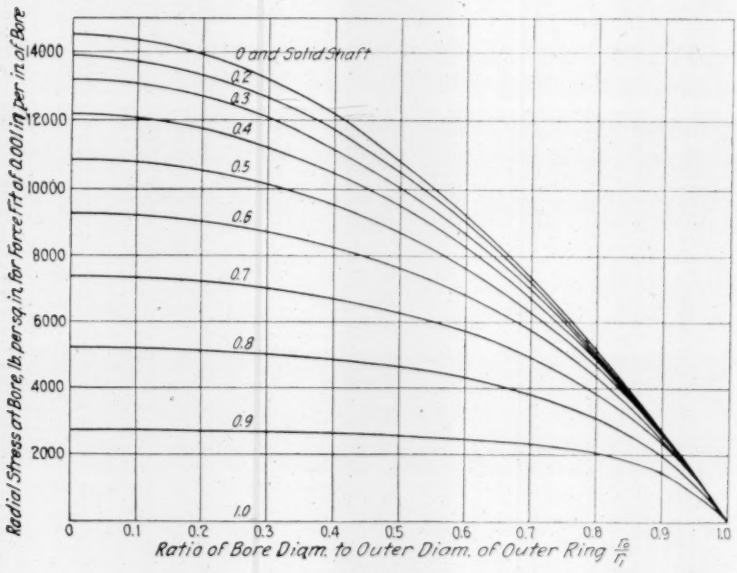

FIG. 5 RADIAL STRESS AT BORE S_{r0}. FORCE FIT OF 0.001 IN. PER IN. OF BORE

Stress is directly proportional to Force Fit per Inch of Bore. Figures on Curves denote Ratio of Diameter of Hole in Inner Ring to Diameter of Bore of Outer Ring $\frac{r_2}{r_0}$.

For a wheel with spokes, we have little more than the hub only. We also estimate an equivalent inner ring if we do not have a solid or hollow shaft. Owing to the comparatively slight influence of the parts of the rings furthest from the fit, a considerable error in estimation of the diameters of the equivalents will cause comparatively small error in the results.

31 Figs. 4, 5 and 6 give amount of expansion and stresses in an outer ring or hub pressed or shrunk on inner ring or shaft.

32 The difference in the free diameters is called the force fit, which we will denote by X. When the hub is in place on the shaft and there is no rotation so as to loosen, it is expanded by a certain amount F'_o. The shaft is also compressed by a considerable amount. The sum of the increase of hub bore, F'_o, and the compression of the shaft is evidently the amount of the force fit. Fig. 4 gives values of $\frac{F'_o}{X}$ the fraction of the force fit which the hub expands.

33 Fig. 5 gives the radial stress at the bore of the hub, which is

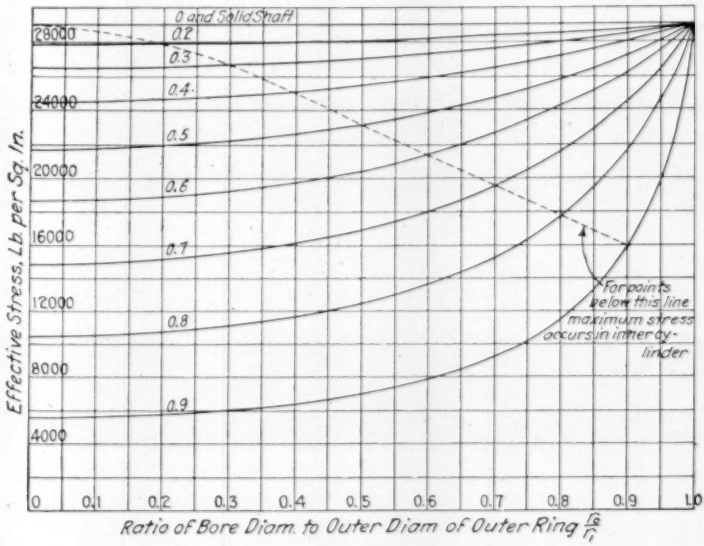

FIG. 6 EFFECTIVE TENSILE STRESS AT BORE OF OUTER RING. FORCE FIT OF 0.001 IN. PER IN. OF BORE

Stress is directly proportional to Force Fit per Inch of Bore. Figures on Curve denote Ratio of Inner Diameter of Inner Ring to Bore Diameter $\frac{r_2}{r_0}$.

also the stress at the outside of the shaft, as well as the pressure per square inch between when there is actual contact for the full length of hub.

34 Fig. 6 gives the maximum effective stress at the bore of the hub. This is the greatest stress in the system unless the shaft is hollow and relatively thin. Figs. 5 and 6 are for a force fit of one mil per inch of bore. Stresses for other values are in proportion.

35 The abscissae of these figures give the ratio of the bore to the outer diameter of the hub. The upper curve in each of these

three figures is for a solid shaft or one with a small hole and the other curves are for rings with various sizes of holes. When the abscissa is zero the outer diameter of the hub is infinitely large as compared with the bore. The curves are nearly horizontal for a considerable distance in this region and the results for an outer diameter ten times the diameter of the bore, that is, a ratio of 0.1, are almost the same as for an infinite outer diameter. The values

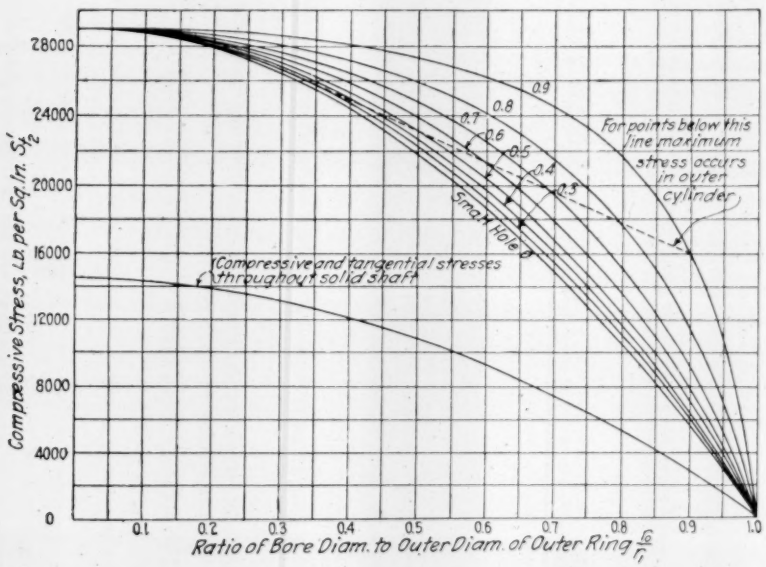

FIG. 7 COMPRESSIVE TANGENTIAL STRESS S'_{t2} AT INNER DIAMETER OF INNER RING. FORCE FIT OF 0.001 IN. PER IN. OF BORE

Stress is directly proportional to Force Fit per Inch of Bore. Figures on Curves denote Ratio of Inner Diameter of Inner Ring to Bore Diameter $\frac{r_2}{r_0}$. Values given are the Maximum Effective Stresses in Inner Ring.

for an outer diameter five or three times the diameter of the bore change but little.

36 Similarly, as will be seen by noting the curves for the various values of the ratio of the diameter of the hole in the shaft to the shaft diameter, the hole has but little influence unless it is comparatively large. The usual case of a shaft with a comparatively small hole is practically equivalent to a solid shaft. As will be seen by a study of the curves, the case where the diameter of the hub is seven times the diameter of the bore, giving a ratio of 0.14, and

where the hole in the shaft is 0.3 of the shaft diameter, gives very nearly the results of any usual case. Hence, for usual cases the hub bore expands about 0.6 of the amount of the force fit and for a force fit of one mil per inch of bore the radial stress is about 13,000 lb. per sq. in. and the effective tangential stress is about 27,000 lb. per sq. in.

37 For a solid shaft, however, the maximum effective stress is 29,000 lb. per sq. in. for a force fit of 1 mil per in. of bore, regardless of the character of the wheel or hub. The stress for other amounts of force fit is in proportion.

38 Force fits of 1 mil per in. are therefore about as large as will give no permanent set of the hub for steel and a solid shaft. Force fits up to about $1\frac{1}{2}$ mils are frequently used. These give some set of the fibers very near the bore. This set is almost entirely eliminated by the elastic force of the outer fibers if the hub is ever removed from the shaft, however.

39 Fig. 7 gives the maximum stress (at the inside) in the shaft or inner ring. This is usually less, but for a relatively thin ring may be greater, than the stress in the hub. A solid shaft has a stress one-half as great as one with a small hole.

40 A rational expression for the force in pounds to press the hub on the shaft is

$$0.12\, d_o l S_{ro}$$

where l is the length, d_o the diameter of bore of the hub in inches and S_{ro} the radial stress at the bore found from Fig. 5. The coefficient of 0.12 was obtained from actual tests and gives a coefficient of friction of 0.038.

41 For a force fit of one mil per inch of bore the forcing pressure in our usual case is about 1560 lb. per in. of bore and per in. of hub length.

42 The forcing pressure of course varies widely on account of the influence of surfaces and lubricant. The formula gives an average value.

APPENDIX

DERIVATION OF FORMULAE FOR FORCE AND SHRINK FITS [1]

NOTATION

S_r = Radial stress, lb. per sq. in.
S_t = Tangential stress, lb. per sq. in.
r = Radius, inches
E = Modulus of elasticity taken 29,000,000 lb. per sq. in.
V = Poisson's ratio, taken 0.3
X = Amount of force fit, or difference in diameters, inches
F = Increase of diameter, inches

40 We take the case of two steel rings, the outer one being forced on the inner one. A subscript after any symbol denotes the value for a particular radius: 1 denotes the outer diameter of the outer ring or hub; 0 denotes the common diameter where the force fit occurs and 2 denotes the bore of the inner ring. We will also use an index $'$ to denote values for the outer ring in general and indexes $''$ to denote the inner ring.

41 The increase in radius of an elementary ring is $\frac{F}{2}$ and the tangential strain is

$$\frac{F}{2r}=\frac{S_t-VS_r}{E}$$

The radial strain is

$$\frac{1}{2}\frac{dF}{dr}=\frac{S_r-VS_t}{E}$$

and by using the preceding value of F we obtain a fundamental equation applying to any circular body.

$$V\left(S_r+r\frac{dS_r}{dr}-S_t\right)=S_t+r\frac{dS_t}{dr}-S_r \quad \ldots\ldots\ldots\ldots\ldots\ldots [4]$$

42 The statement that the radial forces on an elementary segment in a stationary cylinder are balanced is

$$S_r dr\, d\,\theta+r d\,\theta d\,S_r-2\,S_t dr \sin d\frac{\theta}{2}=0$$

This reduces to a second fundamental differential equation

$$S_r+r\frac{dS_r}{dr}-S_t=0 \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots [5]$$

This enables reduction of the first fundamental equation to the form

[1] This section was written with the assistance of Mr. C. A. Schellens of the General Electric Company, Lynn, Mass.

$$S_t+r\frac{dS_t}{dr}-S_r=0 \qquad [6]$$

We integrate these simultaneous differential equations by adding and integrating, giving

$$S_r+S_t=2A$$

We add this to the second fundamental equation which separates the variables. By multiplying by r we obtain the exact differential of r^2S_r so that we obtain the integral directly. From the equation above we obtain the other integral, giving the equations

$$S_r'=A'-\frac{B'}{r^2}$$

$$S_t'=A'+\frac{B'}{r^2}$$

$$S_r''=A''-\frac{B''}{r^2}$$

$$S_t''=A''+\frac{B''}{r^2}$$

43 A' B' A'' B'' are four constants to be determined by the boundary conditions. Two of these are that the radial stresses at the inner and outer radii r_1 and r_2 are zero for the case of a hole in the inner ring. That is

$$S'_{r1}=0$$

$$S''_{r2}=0$$

Substituting these values in the above equations

$$0=A'-\frac{B'}{r_1^2}$$

$$0=A''-\frac{B''}{r_2^2}$$

44 This enables us to eliminate two constants giving a new set of fundamental equations

$$S_r'=B'\left(\frac{1}{r_1^2}-\frac{1}{r^2}\right)$$

$$S_t'=B'\left(\frac{1}{r_1^2}+\frac{1}{r^2}\right)$$

$$S_r''=B''\left(\frac{1}{r_2^2}-\frac{1}{r^2}\right)$$

$$S_t''=B''\left(\frac{1}{r_2^2}+\frac{1}{r^2}\right)$$

45 The third boundary condition is that the radial stresses at the common surfaces are equal, that is

$$S'_{r0}=S''_{r0}$$

giving

$$B'\left(\frac{1}{r_1^2}-\frac{1}{r_0^2}\right)=B''\left(\frac{1}{r_2^2}-\frac{1}{r_0^2}\right)$$

46 The fourth condition is that the algebraic difference between the deformations at the common radius is equal to the force fit. That is

$$F_0'-F_0''=X$$

47 In general the radial deformation at any radius is given by

$$\frac{F}{2}=\frac{r}{E}\left(S_t-VS_r\right)$$

Outward deformation and tensile stress are counted positive.

48 Then we have, after noting that $S'_{r0}=S''_{r0}$

$$\frac{2r_0}{E}\left\{S'_{t0}-S''_{t0}\right\}=X$$

Substituting the stresses from the last set of equations and substituting for B'' from the equation given by the third boundary condition, this becomes

$$B'\left\{\frac{1}{r_1^2}+\frac{1}{r_0^2}-\frac{\left(\frac{1}{r_2^2}+\frac{1}{r_0^2}\right)\left(\frac{1}{r_1^2}-\frac{1}{r_0^2}\right)}{\left(\frac{1}{r_2^2}-\frac{1}{r_0^2}\right)}\right\}=\frac{EX}{2r_0}$$

Hence

$$B'=\frac{EX}{4r_0}\,\frac{r_1^2\left(1-\frac{r_2^2}{r_0^2}\right)}{\frac{r_1^2}{r_0^2}-\frac{r_2^2}{r_0^2}} \qquad [7]$$

This gives a value of B' which we can substitute in the fundamental equations. The value of the radial stress at the common radius then comes out

$$S'_{r0}=-E\frac{X}{4r_0}\,\frac{\left(\frac{r_1^2}{r_0^2}-1\right)\left(1-\frac{r_2^2}{r_0^2}\right)}{\frac{r_1^2}{r_0^2}-\frac{r_2^2}{r_0^2}} \qquad [8]$$

This is a compressive stress. The stress varies in the outer ring from the above value at the bore to zero at the outside.

49 Fig. 5 gives values computed from this equation. Values of radial stress S_{r0} are ordinates, for a value of $\frac{X}{2r_0}$, the force fit per inch of bore, of one mil per inch. Values of $\frac{r_0}{r_1}$ are abscissae. Each of the curves is for a given value of $\frac{r_2}{r_0}$.

50 The fraction of the force fit which the outer ring expands is given by

$$\begin{aligned}\frac{F_0'}{X}&=\frac{2r_0}{EX}\left(S'_{t0}-VS'_{r0}\right)\\&=\frac{2r_0B'}{EX}\left\{\frac{1}{r_1^2}+\frac{1}{r_0^2}+V\left(\frac{1}{r_0^2}-\frac{1}{r_1^2}\right)\right\}\\&=\frac{1}{2}\,\frac{\left\{1+\frac{r_1^2}{r_0^2}+V\left(\frac{r_1^2}{r_0^2}-1\right)\right\}\left(1-\frac{r_2^2}{r_0^2}\right)}{\frac{r_1^2}{r_0^2}-\frac{r_2^2}{r_0^2}}\end{aligned} \qquad [9]$$

51 Fig. 4 gives values computed from this equation. The ordinates are values of $\frac{F'_0}{X}$, the fraction of force fit which the outer ring expands. Values of $\frac{r_0}{r_1}$

are abscissae. Each of the curves is for a given value of $\frac{r_2}{r_0}$.

52 If we substitute the value of B' in the last set of fundamental equations and obtain the value of S'_t we find that there is a tangential tensile stress in the outer cylinder with greatest value at the bore. Hence, since the radial stress at this point is compressive, we have as the greatest value of the effective stress in the outer cylinder, at the bore

$$S'_{t0} - S'_{r0} = E \frac{X}{2r_0} \frac{1 - \frac{r_1^2}{r_0^2}}{1 - \frac{r_2^2 r_0^2}{r_0^2 r_1^2}} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots [10]$$

53 Fig. 6 gives values computed from this equation. For a small hole in the inner cylinder, and also, as will be shown later, for a solid shaft, this maximum stress in the outer cylinder, at its bore, becomes

$$E \frac{X}{2r_0}$$

This value of the maximum effective stress in a hub on a solid shaft, obtained in the correct way by adding the tangential stress due to hub extension and radial stress due to shaft compression, turns out to give the same final results as the incorrect method of finding the tangential stress in the hub with an incompressible shaft.

54 From the relation between B' and B'' and the value of B' we can substitute in the last set of fundamental equations and obtain the stresses in the inner ring. Both stresses are compression throughout so that the greatest gives the maximum effective stress direct.

55 The radial compressive stress increases from zero at the inner diameter to a maximum at the outer diameter of the inner ring, where it has the value already given for the bore of the outer cylinder and shown in Fig. 5. The tangential compression stress increases from the outer diameter to the diameter of the central hole where it is the maximum stress in the inner cylinder and is

$$S''_{t2} = -E \frac{X}{2r_0} \frac{1 - \frac{r_0^2}{r_1^2}}{1 - \frac{r_2^2 r_0^2}{r_0^2 r_1^2}} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots [11]$$

Values of the expression are plotted in Fig. 7. It is to be noted that in case the ratios of radii of both rings are alike, the above expression for maximum stress in the inner ring coincides with the previous expression for maximum effective stress in the outer ring. It can also be seen that the greater of the two maximum stresses is in that ring which has the greatest ratio of inner to outer diameter, i. e., which is relatively thinnest.

56 The preceding work has been for the case where there was a hole in the inner cylinder. For a solid shaft, the inner boundary condition in the inner cylinder becomes

$$S''_{t2} = S''_{r2}$$

instead of

$$S''_{r2} = 0$$

the other conditions remaining the same. From the first set of fundamental equations and the above relation

$$A''r_2^2 - B'' = A''r_2^2 + B''$$

If we now put $r_2 = 0$ we have

$$B'' = 0 \text{ and } S''_r = S''_t = A''$$

That is, the radial and tangential stresses are equal and constant throughout a solid shaft. This is a well-known property of any solid with uniform normal stress over the entire surface.

57 By inserting the remaining boundary condition we obtain final values for the stresses in the outer cylinder exactly the same as the previous values for a hollow shaft when the radius of the inner hole r_2 is made equal to zero giving an infinitely small hole. The value of the uniform stress throughout the shaft is

$$S''_r = S''_t = -\frac{E}{2}\frac{X}{2r_0}\left(1 - \frac{r_0^2}{r_1^2}\right) \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots [12]$$

58 Values of this expression are given by the lower curve of Fig. 7. It is exactly one-half the previous value of the stress at the center of a ring when the radius of the inner hole r_2 is made equal to zero, giving an infinitely small hole.

INVESTIGATION OF EFFICIENCY OF WORM GEARING FOR AUTOMOBILE TRANSMISSION

By Wm. H. Kenerson

ABSTRACT OF THE PAPER

The paper describes a series of tests made to determine the efficiency of three types of worm gearing intended for use in automobile transmission, and at the same time a series of tests to determine the effect of continuous running. Data regarding the shape and construction of the gears are given and the results tabulated and also expressed in graphic form. The results indicate a high efficiency and large carrying capacity.

INVESTIGATION OF EFFICIENCY OF WORM GEARING FOR AUTOMOBILE TRANSMISSION

By Wm. H. Kenerson, Providence, R. I.

Member of the Society

This investigation was made at the plant of the Brown & Sharpe Manufacturing Company for the purpose of determining the efficiency of three types of worm gearing for use in an automobile transmission system and the heating effect due to continuous running. The power for these tests was obtained from a 50-h.p. induction motor, running approximately 870 r.p.m. at full load.

2 Between the motor and the worm gear case was placed an automobile transmission gear case to enable tests to be made at two lower speeds. Between this and the worm gear case was placed a transmission dynamometer designed by the author and described in a paper presented before this Society and published in Vol. 31 of Transactions.[1] An Alden brake was used to absorb and measure the power transmitted by the gears under test. The arrangement of the apparatus is shown in Fig. 1 which shows the transmission dynamometer set up for comparison with the Alden brake, and in Fig. 2 which shows the whole apparatus mounted ready for testing the worm gear efficiency. Referring especially to Fig. 2, *A* is the motor, *B* the automobile transmission gear case, *C* the transmission dynamometer, *D* the case containing the worm gear under test, *E* the Alden brake, and *F* the platform scale which measured the load on the Alden brake.

3 The apparatus was so arranged that as the load was imposed the weight on the platform scale was removed and all vibration of the scale beam was eliminated by interposing springs *S* between the blocks *FF* which sustained the weight on the

[1] Wm. H. Kenerson, A New Transmission Dynamometer, p. 171.

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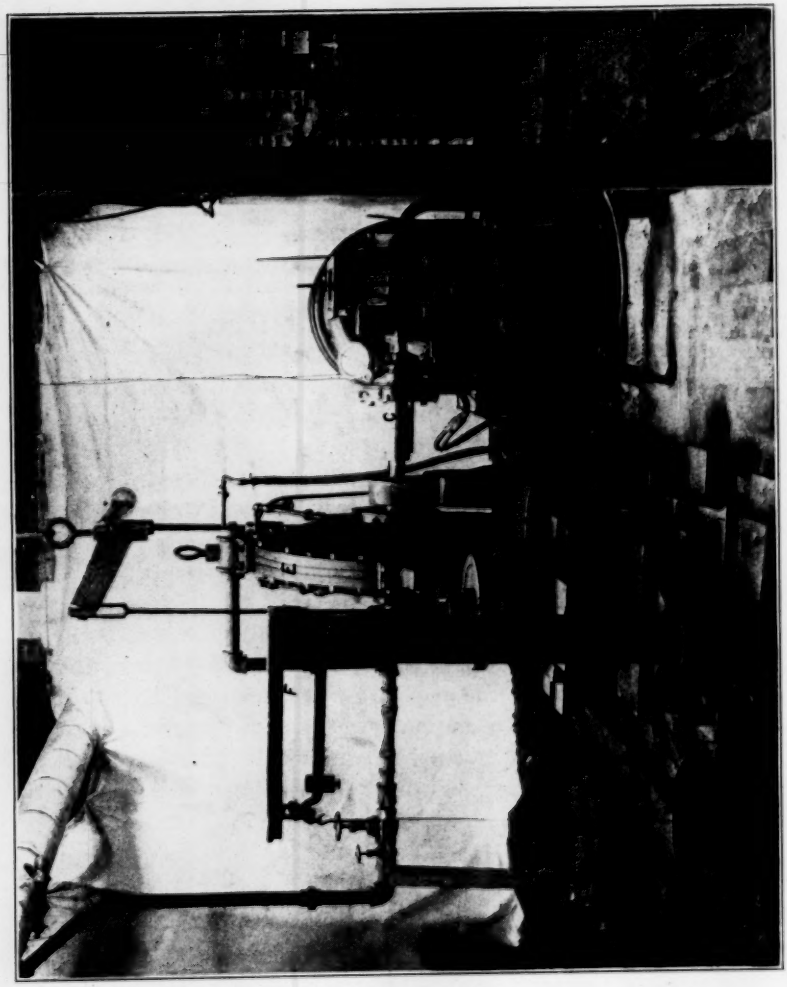


FIG. 1 APPARATUS ARRANGED FOR CALIBRATION

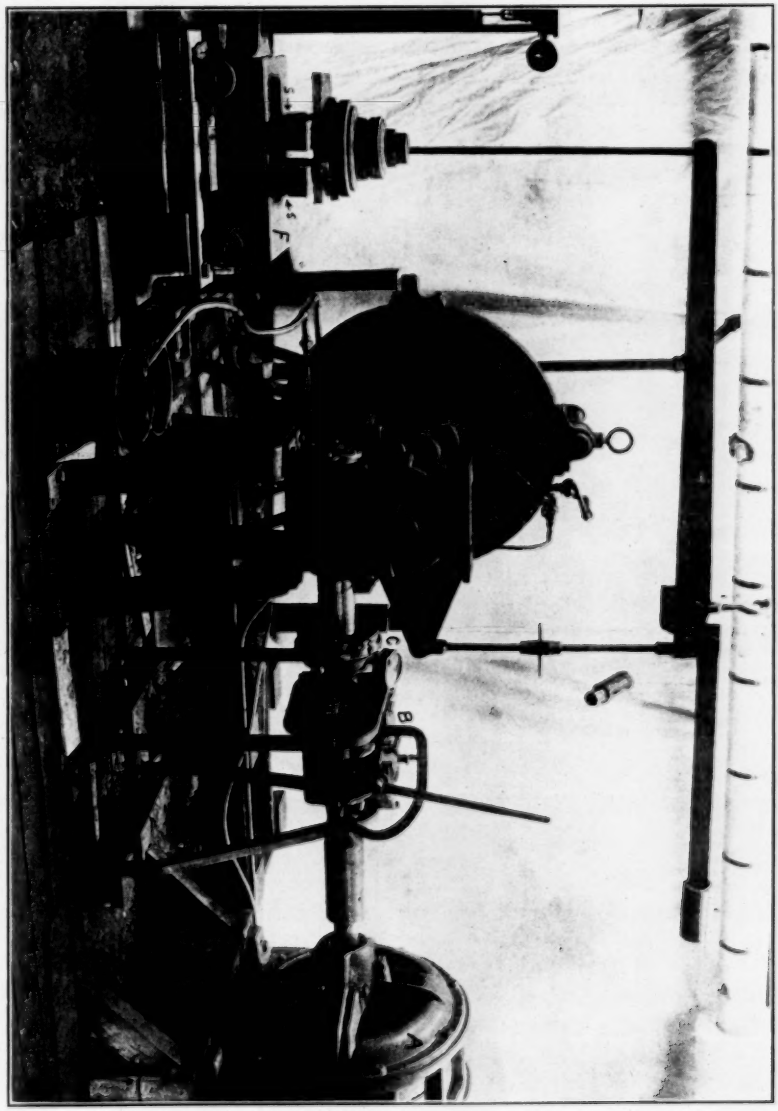


FIG. 2 APPARATUS FOR TESTING WORM GEARS

scale, and also by suspending by a wire from the weight at the end of the scale beam a plate which dipped into a pail of oil, thus acting as an efficient dashpot. It was found possible by careful manipulation to maintain a steady and easily read load on both the transmission and absorption instruments.

4 The transmission dynamometer and brake were first compared by arranging them as shown in Fig. 1, in which position

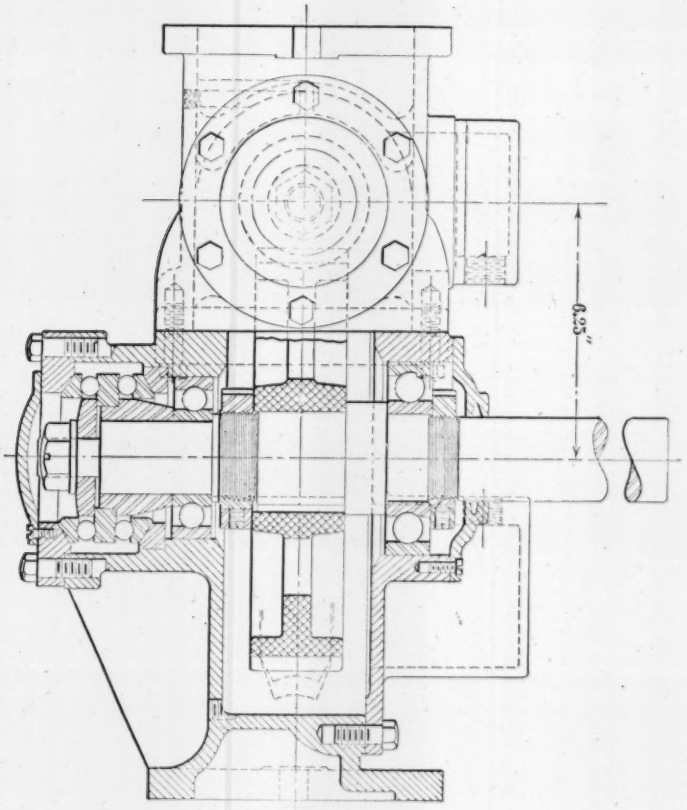


FIG. 3 SECTION THROUGH GEAR CASE

runs were made at various loads, and the torques corresponding to horsepower per 100 r.p.m. marked on the dial of the transmission instrument corresponding to similar loads as indicated on the brake. It is evident that by this method of comparison the two instruments must check each other exactly.

5 A thermometer placed in the oil well at the back of the

worm gear case *D* indicated the temperature of the oil, and another thermometer placed on the wall near the apparatus indicated the room temperature.

6 The oil employed to lubricate the gears was one intended for use with superheated steam, having a specific gravity of 26 Baumé, flash point 625 fahr., and viscosity at 212 deg. 260 to 265. The case contained about 5 quarts of the oil.

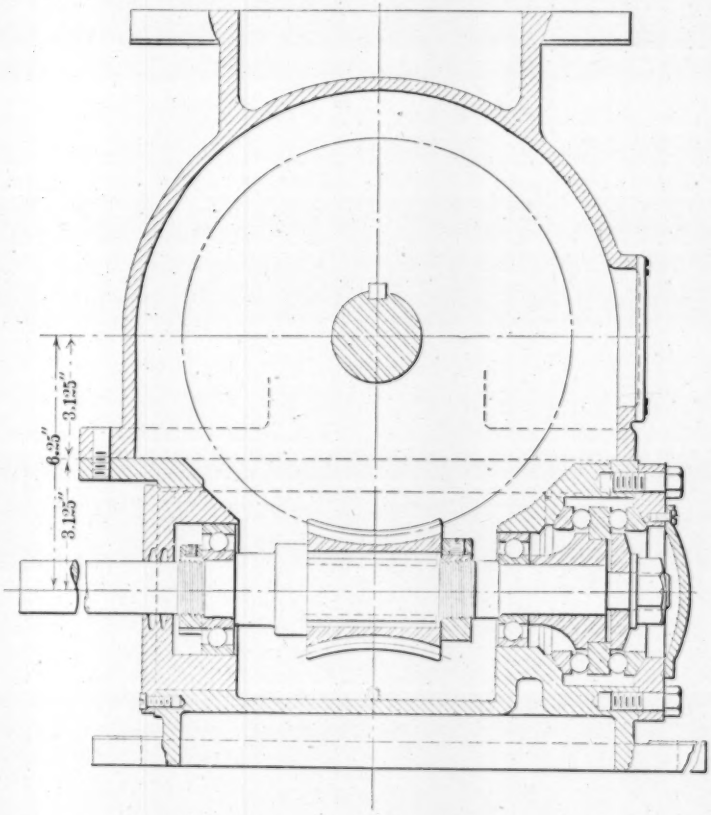


FIG. 4 SECTION THROUGH GEAR CASE

7 In all the trials the worm was located underneath the gear. Figs. 3 and 4 show sections through the gear case. As indicated, both shafts are mounted on ball bearings and end-thrust ball bearings take care of the thrust on the worm and wormwheel. All the worms were made of machinery steel, case-hardened, and the wormwheels of phosphor bronze.

8 The first worm and wheel tested are shown in Fig. 5, and are similar to those used for driving the spindle on the Brown & Sharpe automatic spur gear machines. This pair of gears is an unusual case of the worm and wormwheel. The smaller gear is hobbed with a hob of the size of the larger gear, thus making possible adjustment of the larger gear, which would not otherwise be the case. While in appearance this gear resembles a Hindley worm, it is not such in reality.

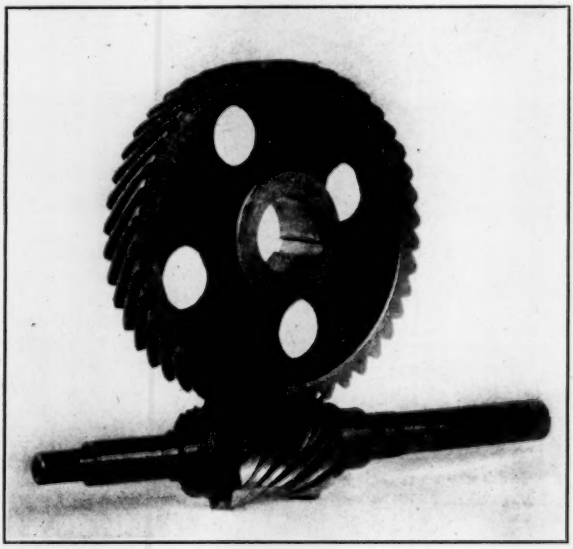


FIG. 5 FIRST WORM AND WHEEL TESTED

9 Although not strictly in accordance with usage, the smaller gear will be called the worm and the larger the wormwheel in the following description. Figs. 5 and 6 give the dimensions and a good general idea of this pair. The wormwheel was cut with the cutter shown, and the shape of the teeth on a section through the worm and wormwheel, parallel to the axis of the worm, is also shown.

10 All necessary information pertaining to pairs No. 2 and 3 is given in Figs. 7, 8 and 9. The difference between gears 2 and 3 is principally one of shape of the worm threads. This difference is clearly brought out in the sections of the worm and wheel shown in Figs. 10 and 11.

11 The shape of the teeth on the worm in Fig. 10 was produced with a cutter, the included angle of which was 29 deg., and the depth of tooth 0.570 in. This depth was based on the axial pitch, when the usual method on multiple worms is to base the depth on the normal pitch. The object in using this cutter was to obtain as many teeth as possible in contact at one time, and also a shape that could be ground with a straight-sided emery wheel.

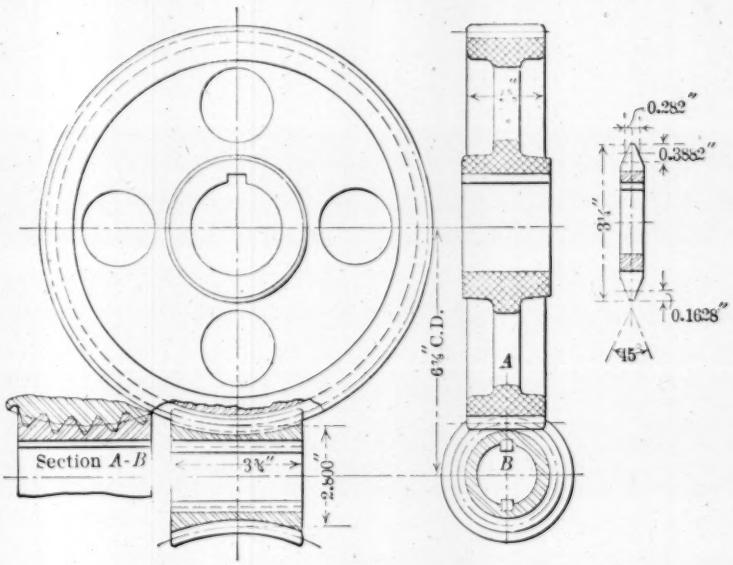


FIG. 6 NO. 1 WORM AND WORMWHEEL

Wormwheel, Phosphor Bronze. No. of teeth, 43 L. H.; Pitch diameter, 10.95; Outside diameter, 11.28; Circular pitch, 0.800; Angle of teeth with axis, 45 deg.; Normal circular pitch, 0.5657; Pitch of cutter, 5.553; Addendum, 0.1628 (not standard); Thickness of tooth, 0.282; Whole depth, 0.3882; Included angle of cutter, 45 deg.

12 The teeth of the worm in Fig. 11 were cut with a cutter shaped as shown, which is an arbitrary shape made to produce at one instant the greatest effective breadth possible, as shown in the figure.

13 In conducting the trials the load was maintained at the desired point by one observer who adjusted the brake. Readings were then taken on the transmission instrument by two independent observers. The speed of the motor was observed in each

case and from this, knowing the gear ratio, all the other speeds were easily computed. Temperatures were taken immediately following each series of observations.

14 For purposes of comparison a series of trials was run on a pair of bevel gears, the principal dimensions of which are shown in Fig. 12. Table 1 is a record of one set of observations typical of the series. Table 2 is a summary of the results of all

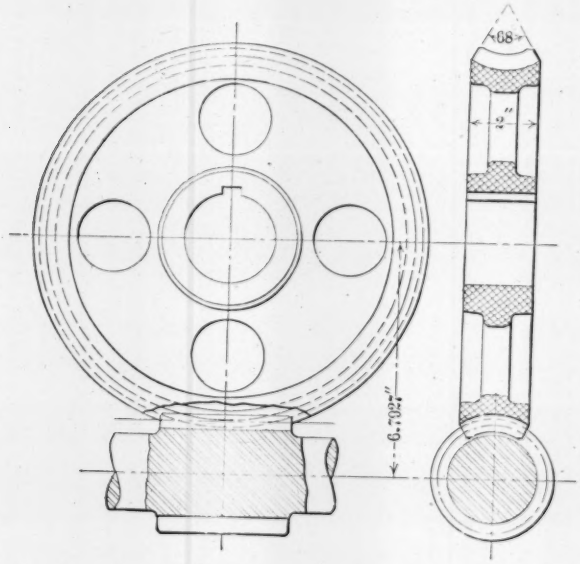


FIG. 7 NOS. 2 AND 3 WORM AND WORMWHEEL

Wormwheel, phosphor bronze; Number of teeth, 40; Pitch diameter, 10.5704; Throat diameter, 10.9964; Circular pitch, 0.8302; Angle of teeth with axis, 38° 16′ 5″; Normal circular pitch, 0.6518; Pitch of cutter, 4.8196; Addendum, 0.213; Thickness of tooth, 0.3568; Whole depth, 0.4586; Worm: Aurora Street, Case-Hardened; Number of teeth, 9; Pitch diameter, 3.015; Outside diameter, 3.441; Circular pitch, 1.0524; Angle of teeth with axis, 51° 43′ 55″; Thickness of tooth, 0.295; Lead, 7.4719; Ratio of Wheel to Worm, 40 to 9.

the trials, and the curves of Fig. 13 show in graphic form the average efficiency of the different gears at the various loads and speeds.

15 In conjunction with the efficiency trials a series of runs was made to determine the heating effect due to continuous running. In these trials, which were in effect endurance tests, a

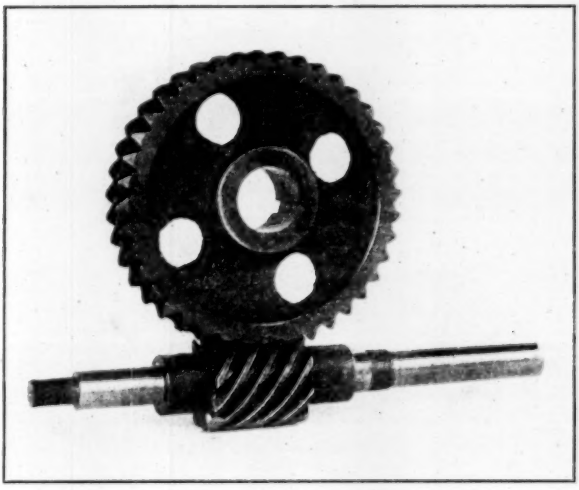


FIG. 8 No. 2 WORM AND WORMWHEEL

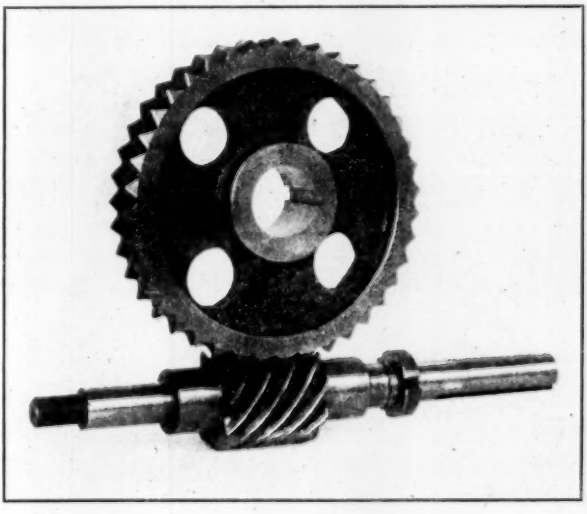


FIG. 9 No. 3 WORM AND WORMWHEEL

TABLE 1 EFFICIENCY TEST OF NO. 2 WORM GEARS WHEN ON DIRECT SPEED

R.P.M. Motor	R.P.M. Worm	R.P.M. Worm Gear	Speed of Worm in Ft. per Min.	Initial Load	Reading on Scale	Load on Brake 315 Ft. Radius	H.P. per 100 R.P.M. Brake	Total Brake H.P.	Total Trans. Dyn. H.P.	Efficiency of Worm Gears	Transmission Dyn. Reading				Temp. at End of Test		Time		Duration
											1	2	3	Av.	Oil	Room	Start	Fin.	
875	875	196.9	690.7	104	55	49	24.5	48.24	49.26	97.9	5.6	5.65	5.65	5.63	268	65	10:14	10:20	6
873	873	196.4	689.1	104	60	44	22.0	43.22	44.35	97.5	5.1	5.05	5.1	5.08	273	65	10:20	10:25	5
880	880	198.0	694.6	104	65	39	19.5	38.61	40.92	94.4	4.7	4.65	4.6	4.65	276	66	10:25	10:30	5
880	880	198.0	694.6	104	70	34	17.0	33.66	36.26	92.8	4.15	4.1	4.1	4.12	275	66	10:30	10:33	3
883	883	198.7	697.0	104	75	29	14.5	28.81	31.17	92.4	3.55	3.55	3.5	3.53	273	66	10:33	10:37	4

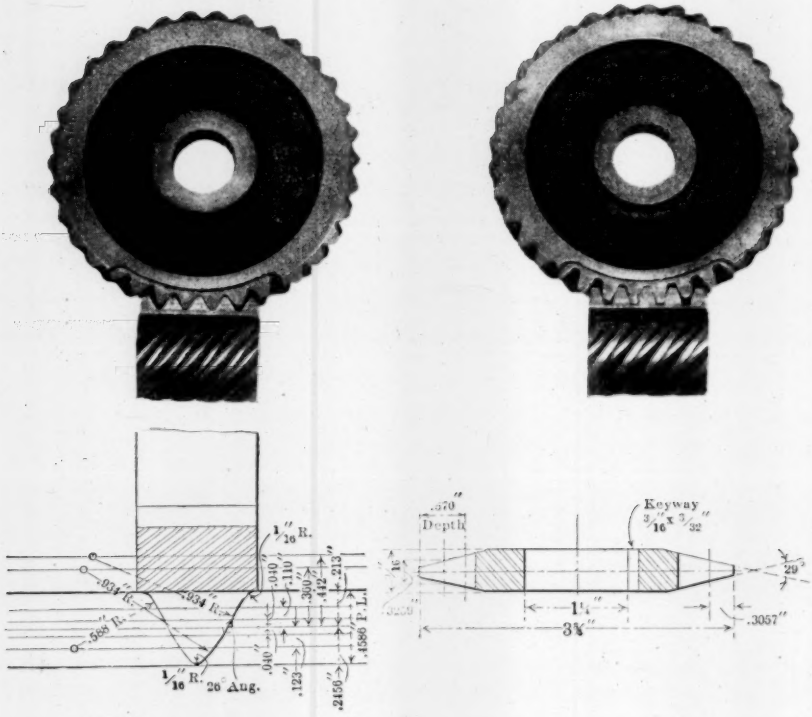
NUMBER OF GROUPS OF READINGS TAKEN IN TABLE 2

	Gear No. 1	Gear No. 2	Gear No. 3	Bevel Gear
First Speed....	4	5	5	3
Second Speed..	4	5	5	6
Direct.....	3	5	5	5

Total Separate Readings, 165

TABLE 2 WORMS AND WHEELS AND BEVEL GEARS, AVERAGE EFFICIENCIES TAKEN

	Gear No. 1							Gear No. 2							Gear No. 3							Bevel Gears						
	R.P.M.		H.P.		Temp.		Eff.	R.P.M.		H.P.		Temp.		Eff.	R.P.M.		H.P.		Temp.		Eff.	R.P.M.		H.P.		Temp.		Eff.
	Worm	Worm Wheel	Input	Output	Oil	Room		Worm	Worm Wheel	Input	Output	Oil	Room		Worm	Worm Wheel	Input	Output	Oil	Room		Pinion	Gear	Input	Output	Oil	Room	
First Speed	254.2	65.03	26.18	24.06	206	73	91.9	254.7	57.32	19.26	18.34	164	69	95.3	254.1	57.19	18.75	18.30	136	79	95.5	254.2	68.43	20.92	20.19		70	96.6
	254.6	65.13	24.75	22.47	217	73	90.8	254.7	57.32	17.99	16.91	169	69	94.1	254.8	57.33	16.54	15.48	144	79	93.1	254.5	68.51	17.50	16.78		70	95.9
	254.5	65.11	22.98	20.81	225	73	90.5	254.5	57.25	16.73	15.46	162	68	92.5	255.0	57.38	13.75	12.62	149	79	91.8	256.1	68.95	14.54	13.44		69	92.4
	255.1	65.25	21.17	19.25	228	73	90.9	254.3	57.22	15.08	14.02	142	69	93.1	255.4	57.47	10.80	9.77	150	79	90.5	256.5	69.05	10.95	10.01		70	91.4
	255.3	65.30	19.56	17.63	232	73	90.1	255.1	57.39	13.52	12.63	159	69	93.5	255.6	57.52	7.38	6.90	149	79	93.6							
	255.7	65.41	17.76	16.02	233	74	90.2	254.9	57.35	12.14	11.18	160	69	92.2														
	255.9	65.48	16.00	14.41	233	74	90.1	254.8	57.34	10.69	9.75	160	69	91.4														
	256.3	65.60	14.34	12.79	233	74	89.4																					
	256.5	65.62	12.37	11.15	231	74	90.2																					
	256.8	65.69	10.55	9.53	229	74	90.2																					
Second Speed	545.2	139.5	43.48	41.16	178	73	94.8	544.2	122.4	36.95	36.11	178	70	97.7	543.2	122.2	38.59	36.06	204	77	93.4	541.8	145.0	39.64	39.37		67	99.3
	546.9	139.9	40.54	37.78	207	70	93.0	545.0	122.6	34.61	33.10	184	70	95.6	543.8	122.4	35.36	33.04	215	77	92.4	544.1	146.4	33.43	32.22		67	96.4
	547.8	140.1	36.88	34.33	214	70	93.2	546.7	123.0	32.09	30.14	188	70	93.9	544.8	122.6	33.12	30.03	219	77	90.7	545.7	146.9	26.25	24.98		67	95.2
	548.8	140.4	33.47	30.89	217	70	92.3	548.4	123.4	28.90	27.15	187	70	93.9	545.8	122.8	29.82	27.02	221	77	90.6	548.5	147.6	18.75	17.72		67	94.5
	550.2	140.7	29.76	27.44	218	70	92.3	549.0	123.5	25.80	24.08	186	71	93.3	546.5	123.0	26.70	23.98	222	78	89.8							
	550.5	140.8	26.01	23.94	218	71	91.3	549.6	123.7	22.59	21.03	163	70	93.1	547.2	123.2	23.56	20.95	222	78	88.9							
	551.1	141.0	22.22	20.45	217	71	92.0	551.0	124.0	19.12	17.98	163	70	94.0	548.3	123.4	20.02	17.89	221	78	89.4							
Direct	859.5	219.8	66.43	64.86	178	69	97.6	873.5	196.5	49.35	48.14	265	66	97.5	866.0	194.8	49.95	47.72	215	77	95.5	872.0	234.7	52.05	51.63		70	99.2
	864.5	221.1	62.89	59.71	201	70	94.9	875.0	196.9	44.50	43.30	256	65	97.3	868.2	195.3	45.68	42.98	223	77	94.1	878.6	236.6	42.09	40.22		70	95.5
	871.0	222.8	59.23	54.60	214	70	92.2	878.0	197.5	40.40	38.50	260	66	95.2	870.8	195.9	41.20	38.20	228	77	92.7	884.4	238.2	30.40	28.58	132	79	94.0
	874.5	223.7	54.22	49.22	224	70	90.8	881.0	198.2	36.40	33.70	261	66	92.4	874.0	196.7	36.88	33.45	230	77	90.6							
	878.0	224.6	48.60	43.80	230	70	90.1	883.4	198.8	30.73	28.84	262	66	93.8	876.0	197.1	31.45	28.60	230	78	90.9							
	882.0	225.6	42.25	38.36	233	70	90.8								878.8	197.7	26.45	23.73	229	78	89.7							
	884.0	226.1	36.24	32.79	237	71	90.5																					



FIGS. 10 AND 11 SECTION OF NOS. 2 AND 3 WORM AND WORMWHEEL

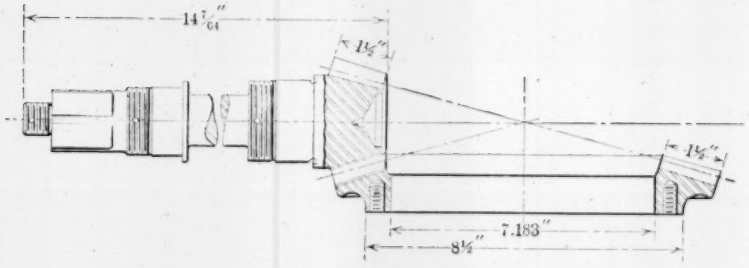


FIG. 12 DRIVING GEAR AND PINION

Driving Pinion, 5 Per Cent Nickel Steel, Case-Hardened; Pitch, 5; Number of teeth, 14; Angle of edge, 15° 4′; Angle of face, 71° 5′; Outside diameter, 3.3359. Driving gear, 5 per cent nickel steel, case-hardened; Pitch, 5; Number of teeth, 52; Angle of edge, 74° 56′; Angle of edge, 13° 47′; Outside diameter, 10.4627.

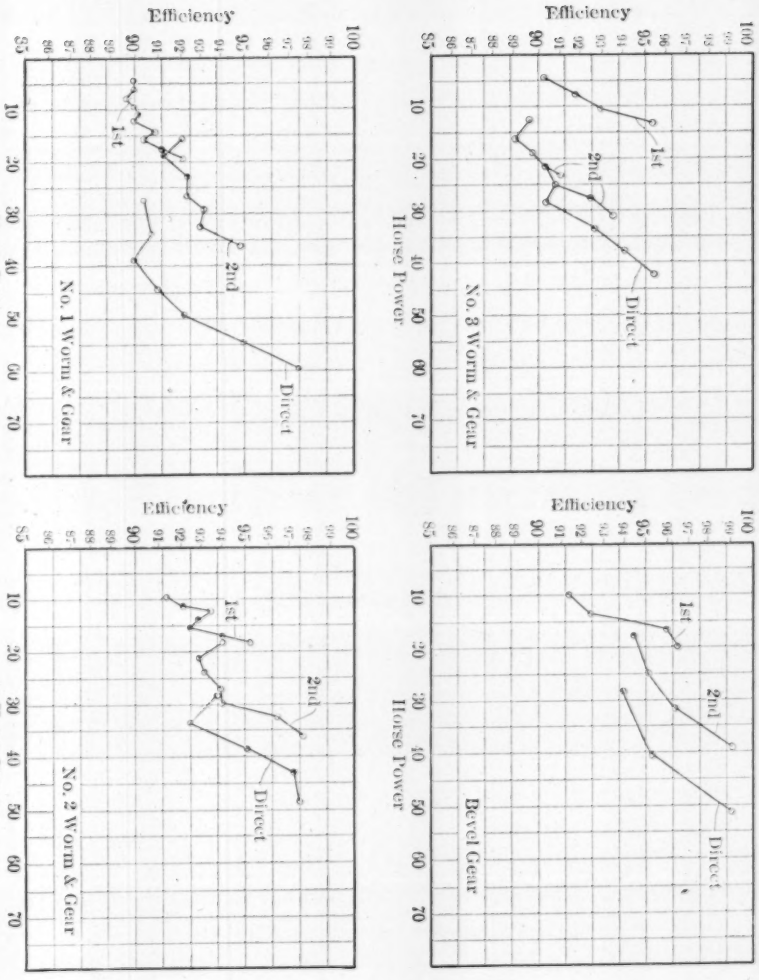


FIG. 13 EFFICIENCY CURVES

constant load was transmitted through the gearing and the temperature of the oil in the gear case and the temperature of the room noted at frequent intervals.

16 From these observations it was found that at the beginning of the run the oil temperature rose rapidly and somewhat irregularly. As the run continued, however, the rise became much more gradual and regular. In the runs where the smaller

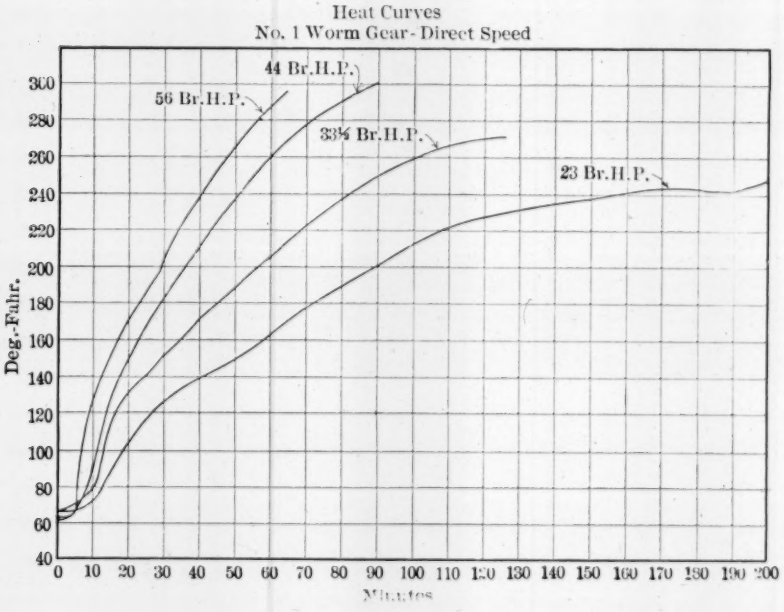


FIG. 14 HEAT CURVES

amounts of power were transmitted a point was reached where the temperature remained constant. This indicated that radiation was sufficient to carry away the heat due to power lost through friction in the gearing, or in other words that the gears would run indefinitely at the load. The heat curves of the No. 1 worm and gear are shown in Fig. 14.

17 The higher loads indicated were abnormal for the gears under consideration and would not occur in any use to which the gears would normally be put. The fact that these trials continued for from 60 to 80 minutes without failure indicates that the structure is both strong and enduring and that should such temperatures be reached for any accidental reason the gears

would not be destroyed. The result of the trials was of particular interest because of the very high efficiency and carrying capacity of the gears tested.

18 Every possible precaution was taken to secure accuracy in the results, and the high degree of accuracy obtained is due largely to the skill and care of Mr. B. F. Waterman of the Brown & Sharpe Manufacturing Company, under whose personal supervision the apparatus was erected and all the trials were conducted.

MEASUREMENT OF AIR IN FAN WORK

By Charles H. Treat

ABSTRACT OF PAPER

The paper discusses the use of the pitot tube for fan testing with particular emphasis on the correct kind of tube to be used. It describes a simple and accurate method of calibration of impact and static orifices, and the instruments to be used in fan testing, as well as the general method and arrangement of the tests. It shows further the derivation of the formula for friction in test pipe, and the method of correcting test results to standard weight air. The paper further contains proofs that the volume discharged from a fan is the same at all densities, other conditions being the same, and indicates the formulae and calculations used in the test, such as the formula for the determination of air horsepower on the discharge and suction side, efficiency of a fan and its speed. Finally, the question of the average velocity of flow in a pipe is discussed.

MEASUREMENT OF AIR IN FAN WORK

By Charles H. Treat, Honor, Mich.

Junior Member of the Society

Air at fan pressures has come to be used in many fields and for widely different lines of work. For the many occasions where measurement of volume, or of pressure and volume, are required better and more accurate methods are often desired than have been commonly used. The engineer frequently needs something better than the ordinary anemometer to determine the velocity, and a rubber tube connected to a water gage to get the pressure. Especially is this so in fan testing where the volume varies from nothing with closed gate to a maximum with free discharge. The measuring apparatus must be accurate through this wide range.

2 The pitot tube[1] used with suitable pressure measuring devices is particularly well adapted for measuring both pressure and volume, being easily and conveniently arranged for fan testing, also being simple, reliable and accurate.

3 Although the pitot tube has long been known and used there is still much doubt of its accuracy and distrust of results obtained by it. This is especially so for air. The writer believes the trouble in some cases and the resulting general distrust are due to a lack of information as to what forms can *not* be used. In this paper are described what is believed to be an accurate form of tube for this purpose and a sufficiently simple way of using it. The pitot tube will be the only means[2] described, although much of the method employed in the test can be used however the values of pressure and volume were found.

4 This pitot tube work was done from time to time in the Amer-

[1]For references and a good account of what has been done see The Pitot Tube, W. B. Gregory, Trans. Am. Soc. M. E., vol. 25, p. 184.

[2]Professor Thomas of the University of Wisconsin has developed an excellent device for measuring air or gases known as the Thomas Meter.

The American Society of Mechanical Engineers, 29 West 39th Street, New York. All papers are subject to revision.

ican Blower Company's fan testing room in Detroit. A little turbine fan was being experimented with by the writer who wanted something smaller and better adapted for use in the small test pipe than the form of tube previously used.[1] The form of pitot tube adopted and which has been in almost daily use for a number of years is shown in Fig. 1. This has been tried out in many ways as by measuring the same air through pipes of different diameter; by measuring in the same pipe with two or more instruments; by measuring the air displaced by a gasometer. These three checks bring in also the

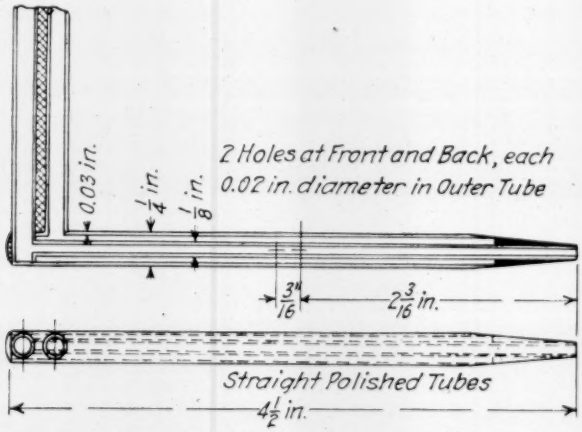


FIG. 1 STANDARD PITOT TUBE

important factor in air measuring of getting the mean velocity in the pipe.

CALIBRATION OF IMPACT AND STATIC ORIFICES

5 In deciding how experimentally to prove the correctness of the orifices of the pitot tube, preference was given to the method which seemed most simple and accurate and which promised to be the best check on work already done.

6 The accuracy of the impact orifice can not be proved by means of a jet of air because the only practical means for determining the velocity at any point in the jet is by the use of the pitot tube. It would scarcely do to test the tube by means of a carrier, such as a locomotive, because of changing air currents. But by attaching the impact tip to the end of a slender hollow arm rotating about a shaft, uniform motion may be obtained and that in air having motion due only to the slender arm through it. If now the shaft be a fan

[1]The pitot tube previously used was the static orifice shown in Fig. 11 at *D*, used with a $\frac{5}{16}$-in. tube pointing into the air current for impact orifice.

shaft and the fan be discharging, air is drawn to the inlet about which the arm is rotating, and by regulating the fan discharge, air can be just drawn away from the path of the tip or, in other words, air flow will be towards the inlet and at right angles to the tip travel instead of with it.

7 We have here a simple means to compare the pressure computed from the relative velocity of tip to air, by the relation $v = \sqrt{2hg}$, with the pressure actually resulting from this velocity. We deal

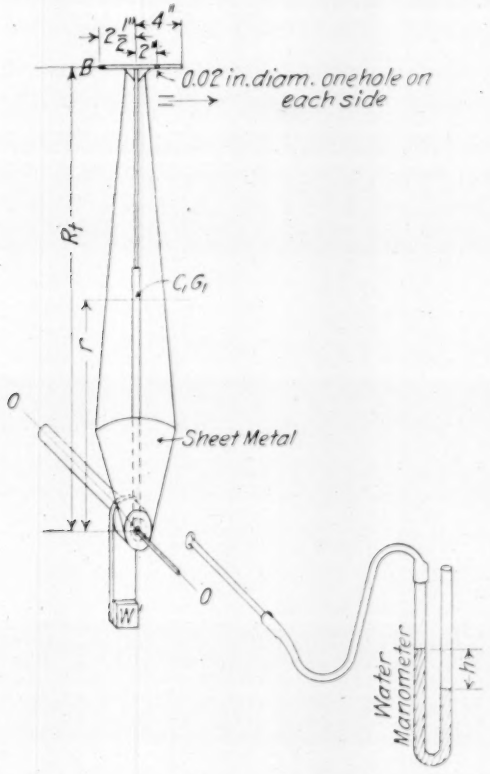


FIG. 2 DIAGRAM FOR APPARATUS FOR CALIBRATING PITOT TUBE

with pressure due to impact only as the atmospheric pressure in the room is on the tip and pressure measuring instrument alike.

8 This rotating arm method was the one used for calibrating not only the impact orifice but the static orifice as well. For the static orifice what we wish to know is whether or not it shows the true pressure of the fluid no matter what the velocity may be. As there may be slight radial flow static openings had best be put at the sides (Fig. 2) for then this velocity is only a component of the total veloc-

ity relative to the static opening. If the pressure in the tip be the same as that in the room the pressure at the axis will be less by the centrifugal head in the arm.

9 If the fan be not used the flow caused by the rotating arm is enough to reduce the pressure in the tip from about 2 to 5 per cent, while the component of flow toward the inlet when the fan is used even with the tip pointing to take the resultant maximum velocity will change the pressure less than will show on a slide rule, flow towards the inlet being only enough to take the air away.

10 In older pitot tube work total or impact reading has usually been all right, but static readings have in many cases been inaccurate. A very simple and reliable check on static openings, but which may not be quite accurate, is to try them in a jet of air blowing out freely

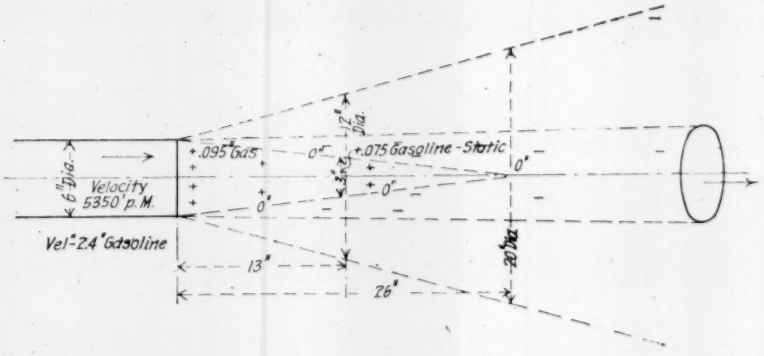


FIG. 3 STATIC PRESSURE AT OUTLET OF A STRAIGHT PIPE

into the room and observe if the pressure is the same as that in the room, as it should be very nearly even with high velocity. Fig. 3 shows what the static pressures were at the end of a straight 6-in. diameter pipe, the velocity pressure being 2.4 in. head of gasolene, equal to 1.74 in. of water.

11 For details of the calibration work, method of computing the centrifugal head, data of tests, etc., see the Appendix. The results are as follows: Working in air instead of water, it is shown that there is no difficulty in getting correct impact or total readings. The form of the tip may be changed from a straight open tube pointing into the current to a small pointed orifice, as in Fig. 1, or may be flat at the tip with a small orifice in the center and still show practically no difference in the readings for the different shapes, showing

that for any ordinary change in the tip no inaccuracy results. The form of impact tip most carefully checked was that shown in Fig. 1.

12 The calibration shows the impact or total reading to be accurate probably to within 1 per cent in the relation $v = \sqrt{2gh}$.

13 A correctly reading static orifice is not so easily obtained. The static of Fig. 1, consisting of clean holes 0.02 in. in diameter in $\frac{1}{4}$-in. tubing $\frac{1}{32}$ in. thick, gave static readings accurate to within less than 1 per cent of pressure due to velocity. A hole $\frac{1}{16}$ in. in diameter in this tubing gave readings considerably off, while a $2\frac{1}{2}$-in. slot $\frac{1}{16}$ in. wide in slightly larger tubing gave static readings as much as 14 per cent too high.

14 An older form of static orifice, shown at *D* in Fig. 4, is fairly accurate, but requires to be set with the plate exactly parallel with

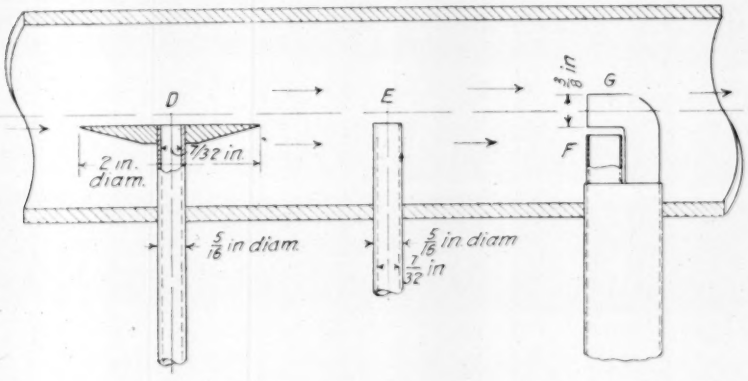


FIG. 4 OLDER FORMS OF STATIC TIPS

the direction of flow. It is often not known just what the direction of flow is, as when there is a slight whirling or spiral motion in the duct. The static openings of the pitot tube shown in Fig. 1 give quite accurate results when 5 deg. out of parallel with flow.

15 The open tube *E*, Fig. 4, gives static readings too small by over 50 per cent of the velocity reading. The tube is $\frac{5}{16}$ in. in diameter held at right angles to flow. The static tube *F* in a form of pitot tube that was sold by one of the instrument makers several years ago gives static readings too small by 65 per cent of the true velocity reading. The impact tube gives practically true reading, so that velocity readings by this instrument are much too large.

16 A $\frac{5}{8}$-in. slot 0.01 in. or less wide in the $\frac{1}{4}$-in. tube seemed to give good results with or without inner tube, so also did the large

slots if an inner tube fitted the outer to within 0.01 in. on each side. Wire cloth over the large slots seemed to give good results. These other forms of opening have not been checked and rechecked as have the small 0.02-in. holes. Where the small holes are liable to stoppage, more holes should be used.

METHOD OF TESTING FANS

17 When air is discharged into a pipe by a fan both static and velocity pressures measured from atmospheric are positive, and their sum is shown by the reading in the impact manometer (*B* in Fig. 5). When the air is drawn from a pipe by a fan the static pressure in the pipe is always less than atmospheric; that is, it is negative. The reading in the impact manometer *B* will be also

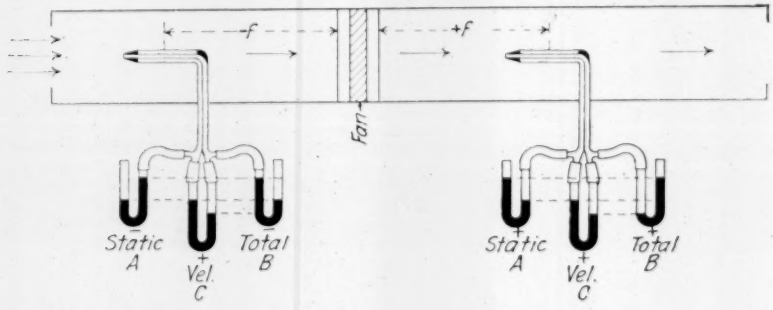


FIG. 5 ARRANGEMENT FOR MEASURING STATIC AND VELOCITY PRESSURES

negative, but less than static. Static subtracted from the impact or total gives a positive reading, as it should, for the velocity. As velocity in this pipe is induced by the static pressure, the velocity can never quite equal the static due to the presence of friction.

18 Figs. 6 and 7 show the method of setting up for a fan test: Fig. 6 for a test on the suction and Fig. 7 for a test on the discharge. Fig. 8 is from a photograph showing the actual arrangement of apparatus under test.

19 By using the pitot tube shown in Fig. 1, where velocity and static are measured practically at the same point, and piping as at *C*, Fig. 5, the pressures other than that due to velocity cancel each other. The possibility of the pressure changing between times of reading static and impact independently is avoided and there is no subtraction to be made. The difference in static pressure between the point of reading in the pipe and atmospheric pressure is read at

A, the total at *B*. The pitot tube should point straight into the current of air to read correctly, either velocity or static.

20 The common pressure measuring apparatus for pressures above 1 in. water gage is satisfactory. For pressures from 1 in. to 2 or 3 ft. of water, manometers are reliable and convenient. Various forms of apparatus are used for measuring pressures of 1 in. or less. As simple and reliable a means as any is probably furnished by an inclined manometer containing gasolene instead of water. The tubes for this manometer should be of the same size, perfectly straight and parallel inside. When so made they can be easily used when inclined as much as 20 in. slant for 1 in. vertical reading, with very accurate results. Water is not a satisfactory fluid in a manometer inclined more than three to one. Gasolene is recommended, as it is not only very active but comes accurately to the same form of surface.

21 Set the back plate by leveling by bubbles in the manometer tubes and then swing the tubes up the angle required to give a good reading, 20 or 10 or 5 to 1. In fan testing for velocity reading at different restrictions two or three slants may be used to cover the wide range in pressure and it is very convenient to cover this wide range with the one gage by the simple device of changing the angle. Good care is necessary to get the angle correct for 20 to 1 and great care for 40 to 1.

22 At 40 to 1 slant for $\frac{1}{2}$ in. velocity reading the velocity is 384 ft., for 0.49 in. reading the velocity is 380 ft., showing that with the slant gage velocities as low as 500 or even 400 ft. per min. may be measured satisfactorily.

23 With skill and care the hook gage gives good results if the pressures do not fluctuate. In air measuring work the pressures usually fluctuate slightly even when care is taken to get the speed of fan as constant as possible. It is desirable to be able to take the readings under these conditions, and this may be done by setting zero-point of the measuring slide at the mean lower level, and at the instant the level comes opposite this point, reading at the upper level.

24 The spacing board for the pitot tube should be placed so that the tube crosses the test pipe centrally. On this board lay off *AB* at the center of travel of the holding block *K* for the tip at extreme movement across pipe. Compute *a*, *b*, *c*, *d* and lay off with *AB* at their center. Inclining manometers are shown for pressure measuring, but other instruments for this purpose may be used if

preferred. After connecting try out the lines of tubing by noting if the liquid lowers in the manometers after pressure has been applied, and make sure that the connections are not clogged. Use care to prevent liquid from being drawn from the gages into the tubing.

25 Make the area of the test pipe the same as the area of the outlet or inlet connections for a fan test. Use well fitted soldered pipe for the test. An ordinary test pipe unsoldered and somewhat carelessly fitted to the fan may leak 5 per cent or more of the air

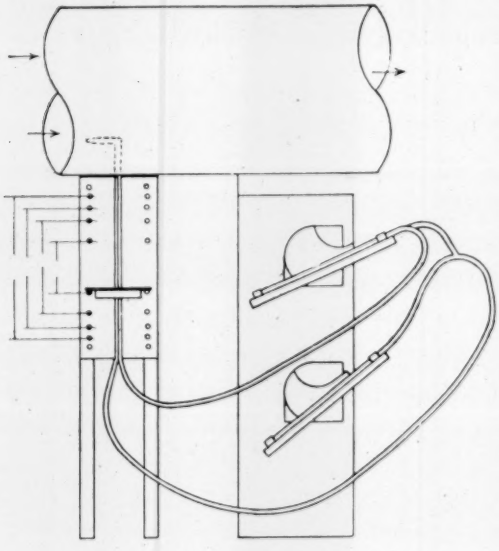


FIG. 6 CORRECTION FOR TEST ON SUCTION

handled at $\frac{1}{2}$ gate, or even at $\frac{5}{8}$ gate, lowering the pressure correspondingly.

26 It is very essential to determine the relation between volume, the pressures and the power required as the resistance changes. For this purpose it is usual to gate down the outlet for an outlet test or the inlet for an inlet test, using round orifices in thin flat plate. No air should escape except through the orifice. It is usually ample to test with gates 0, $\frac{1}{4}$, $\frac{3}{8}$, $\frac{1}{2}$, $\frac{5}{8}$, $\frac{3}{4}$, and once the area of the orifice. Occasionally a $\frac{7}{8}$ gate is useful. For some tests the $\frac{3}{8}$ and the $\frac{5}{8}$ gates may be omitted. The area of the gate divided by the area of the test pipe is the ratio of opening or the equivalent orifice.

27 The coefficient of efflux is larger for a conical outlet than for an orifice of the same area and is probably greater for a large gate that for a small on the same test pipe. However, a gate, say $\frac{3}{4}$, is very nearly a fixed measure of resistance as above made. Even if it were not there would be no inaccuracy due to this cause as the final curves are ordinarily used, for the *required* ratio of opening is found from the *required* ratio of pressures. The ratio of gate area to full area is often very useful, so that it is wiser to use ratio of opening for abscissa in final curves than some other quantity

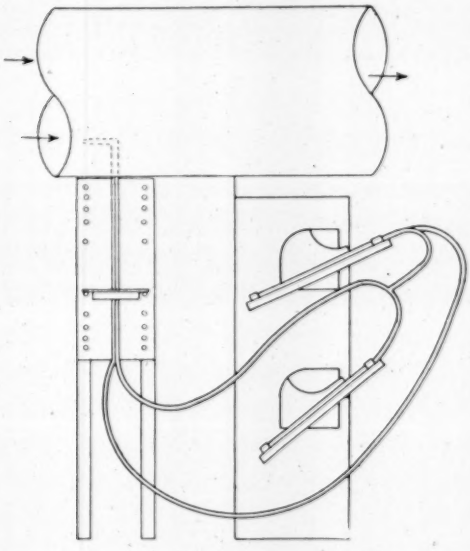


FIG. 7 CORRECTION FOR TEST ON DISCHARGE

as volume or ratio of $\frac{V.P.}{D.P.}$

28 A convenient log sheet for fan testing is shown in Fig. 9. On this log and on the curve sheets, Figs. 9 and 10, pressures are all expressed in inches of water. The pressure in the impact tube or the total or dynamic pressure is marked Dyn. *W. G.* or simply *D.P.* The pressure in the static orifice (*A* in Fig. 5) is Static *W.G.* or simply *S.P.* The difference between the static and the total (*C* in Fig. 5) is the velocity pressure and is marked Vel *W.G.* or simply *V.P.* The pressure due to a velocity of the air equaling the peripheral velocity of the wheel is the peripheral velocity pressure or *P.V.P.*

29 In making fan tests the speed may not be constant at all times. To correct to a common speed we have relations between speed and pressure and power that have been fully verified by test, the better the tests the better being the check. See curve sheet *A* for a good test showing this point. These relations are reasonable and could probably be established by theory. They are as follows:

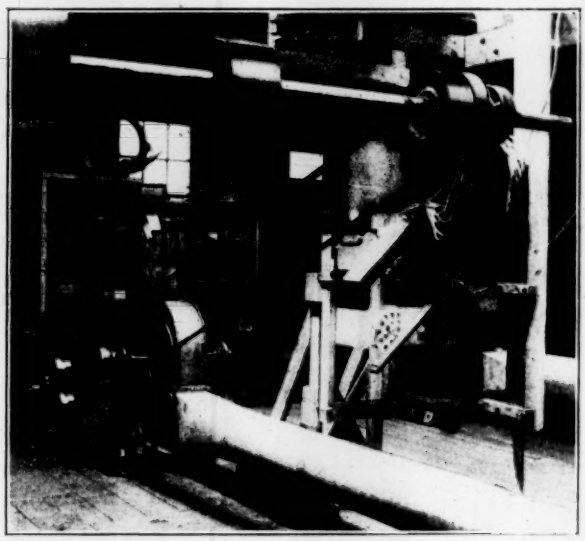


FIG. 8 FAN TESTING

30 Under the same conditions of resistance to air flow, density of air and with the same fan

a The volume of air discharged by the fan varies as the r.p.m., that is

$$\frac{\text{vel.}}{\text{Vel.}} = \frac{\text{r.p.m.}}{\text{R.P.M.}}$$

b The pressures produced vary as the square of the r.p.m.

$$\frac{\text{static}}{\text{STATIC}} = \frac{\text{total}}{\text{TOTAL}} = \frac{\text{velocity}^2}{\text{VELOCITY}^2} = \frac{\text{r.p.m.}^2}{\text{R.P.M.}^2}$$

c The horsepower to drive varies as the cube of the r.p.m.

$$\frac{\text{h.p.}}{\text{H.P.}} = \frac{\text{r.p.m.}^3}{\text{R.P.M.}^3}$$

31 *Friction in Test Pipe.* In working up a fan test, a correction for friction in the test pipe must be made to show what the fan alone

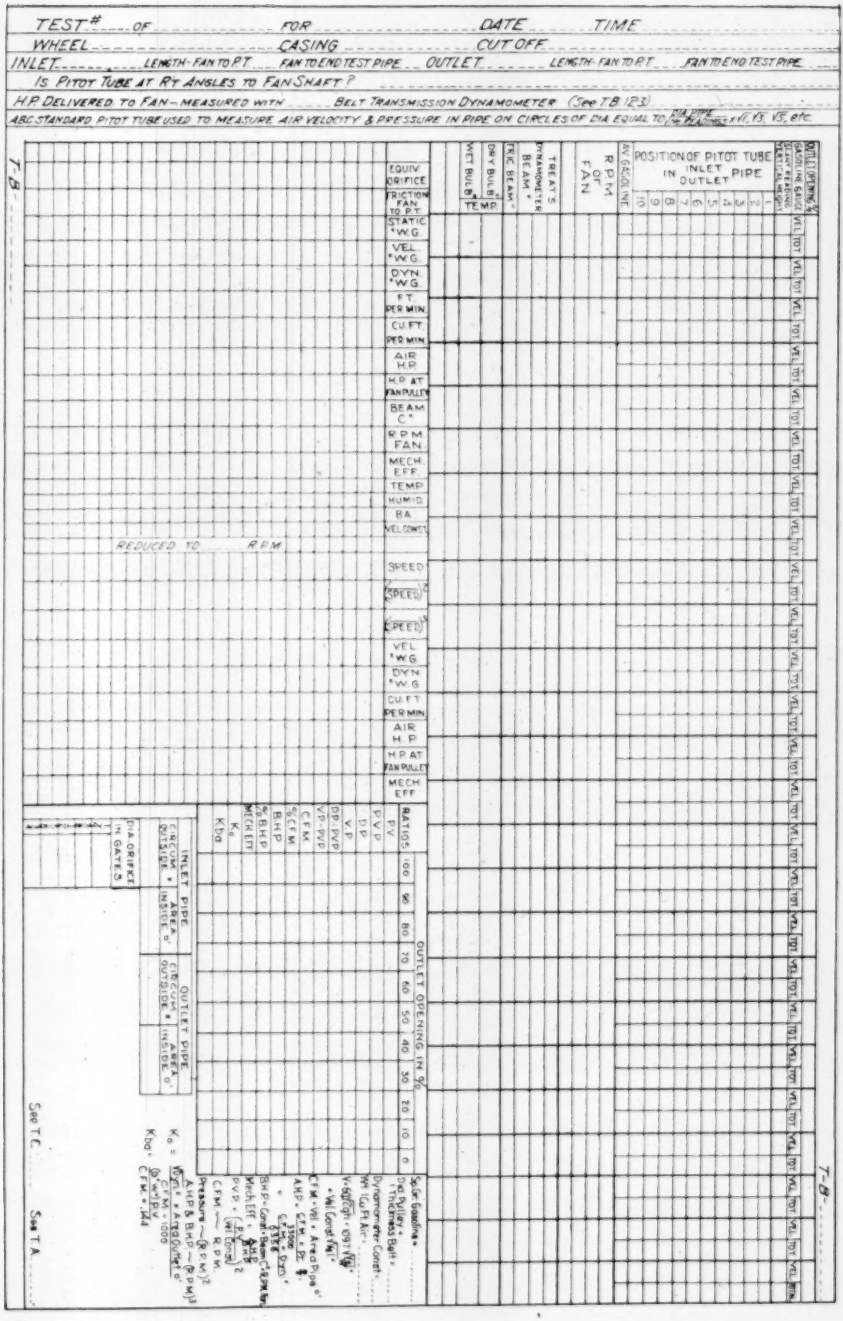


FIG. 9 LOG SHEET FOR FAN TEST

is really doing. The friction pressure or pressure to overcome this friction may be 10 to 15 per cent of the total at wide open gate and 3 to 4 per cent at ½ gate.

32 For ordinary round galvanized pipe a simple and quick way to find this friction is from the formula

$$F = \frac{\text{Vel.}'' \times L'}{40 \times D'} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots [1]$$

where F = friction in inches of water L and D = length and diameter of round pipe. Or in a form more easily remembered, the length of pipe in which friction inches equal velocity inches pressure is 40 diameters long. This rule is applicable for air under ordinary conditions of temperature, pressure and humidity, and values by it check closely with those given by Kent.

33 For smooth straight pipe instead of 40 times the diameter it may be 60 or more and for rough beaded pipe the figure will be less that 40.

34 A general equation[1] for friction is

$$P = \frac{fsyv}{A2g}$$

for any pipe, and

$$P = \frac{4\,fyLv}{2gD} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots [2]$$

for round pipe

y = weight of 1 cu. ft. of air
P = pressure lb. per sq. ft.
L and D = length and diameter of pipe in ft.
f = coefficient of friction
v = velocity in ft. per sec.
s = perimeter of section of pipe

41 If we substitute in [2] the value 0.0067, that being the coefficient of friction in ordinary piping, for coefficient of friction and 0.0715 as the weight of air we can derive [1]. It is interesting to note that 0.0067 is given as the coefficient of skin friction for water of rather rough surface. Is the coefficient for air and water the same? Why should they not be the same?

42 *Standard Weight Air.* It is important to have a standard weight of air at which fan performances may be compared. The density, while ordinarily a small factor, affects the results consider-

[1] See Church's Hydraulics.

ably, if extremes are great, as with heated air or with cold out-door air. It affects the velocity pressure, the total pressure and the horsepower.

43 As correction to a standard involves some labor it is desirable to choose for a standard a mean weight so that the original data will be close enough without correction in many cases, or so that the conditions may be kept such as will give standard weight. Another reason for the mean or ordinary weight is that such data as taken may be compared with previous tests, also that ordinary rough uncorrected tests may be compared with standard tests.

44 As fan tests are apt to be conducted indoors with warmed air the weight on the average for the year in the United States will not be far from 0.0715 lb. per cu. ft. This in fact is the average weight for a large number of tests extending over a number of years. This weight gives a velocity constant of 4100 or

$$\text{vel.} = 4100 \sqrt{\text{vel.}''}$$

There is some advantage in a short figure in computations from the final curves.

45 To correct to standard air, multiply the velocity pressure, total pressure and the horsepower of the actual test by $\frac{0.0715}{\text{wt. 1 cu. ft.}}$ The velocity will be the same as though found at the actual velocity pressure, but the static and total pressure change as the density, changing the air horsepower and the actual horsepower in proportion.

45 A fan discharging under constant conditions as to speed and resistance, will maintain a constant head and a constant velocity of discharge with changing density. That this is so is apparent from the following:

47 In a pipe system supplied by a fan running at a constant speed all the resistance may be classed as friction of various kinds. The impeller sets up heads, due to centrifugal force and to motion imparted to the air, which total head is resisted by internal and pipe frictions.

$$\text{The net head } h_o + h_v + h_f + h_{fi} = \frac{V^2}{2g} \quad \ldots\ldots\ldots\ldots [3]$$

48 Density of the fluid does not affect these heads, and the net head for one density will be the head for all. The velocity will be the same for all from the law of falling bodies.

Centrifugal head

$$P = h_c y = \frac{wv^2}{gr} = \frac{2ryv^2}{gr} \quad \ldots\ldots [4]$$

$$h_c = \frac{2rv^2}{gr} \quad \ldots\ldots [5]$$

Head due to motion

$$h_v = \frac{v^2}{2g} \quad \ldots\ldots [6]$$

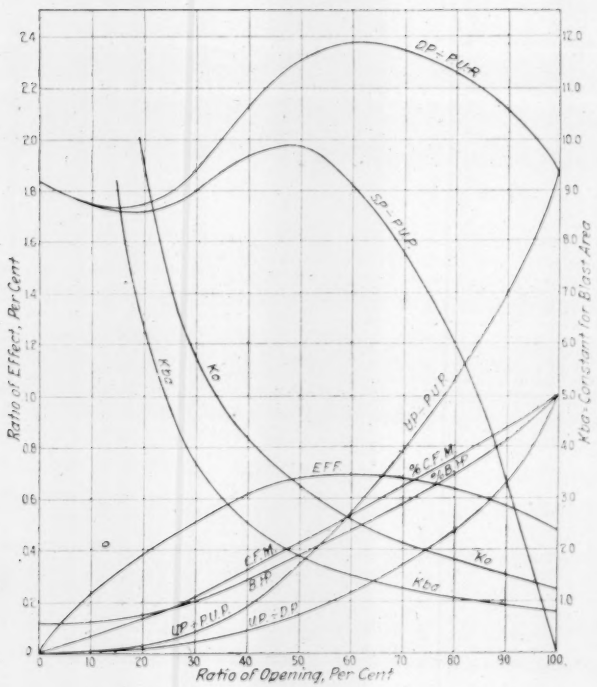


FIG. 10 CHARACTERISTIC CURVES OF A FAN TEST

Heads due to friction pressure

$$P = h_f = \frac{fSyv^2}{A2g} \quad \ldots\ldots [7]$$

and

$$h_f \text{ or } h_{f1} = \frac{fSv^2}{A2g} \quad \ldots\ldots [8]$$

h_c = head due to centrifugal force.
h_v = head due to velocity and velocity change

h_f = head due to friction in case
h_{f1} = head due to friction in pipe
P = pressure per sq. ft.
y = density or weight of 1 cu. ft. air

49 From this it follows that the volume discharged from a fan is the same at all densities, other conditions being the same. This is probably true for all fluids, since the coefficient of friction seems to be the same.

50 A thoroughly good way of making the results of a fan test available and useful is to plot characteristic curves, as in Fig. 10.

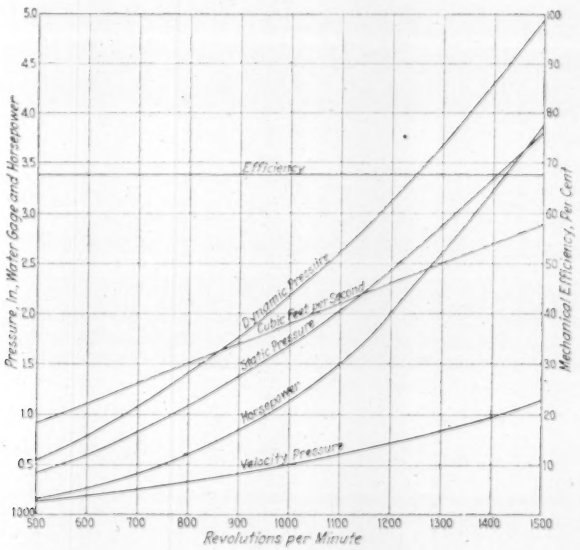


FIG. 11 PRELIMINARY CURVES OF FAN TEST

51 The original readings are first corrected to one common speed, if the speed has varied through the test, as it is likely to do from belt slip. These data at a common speed are then plotted on a preliminary curve sheet similar to the curve sheet in Fig. 11, but with the ratio of opening plotted as abscissa instead of r.p.m. On this sheet correction can be made to standard weight air and for pipe friction. This preliminary set of curves will usually show up a mistake in observation or in computation. From this curve sheet practically any fan problem may be solved for fans of that kind.

Of course the other fans of the line must be kept in proportion to the one tested to get the result.

FORMULAE AND CALCULATIONS USED IN TESTS

52 Bernouilli's theorem states that the sum of all the pressures of a fluid at one section of a system of smooth piping equals the sum of all the pressures at any other section, pipe friction being cared for by the difference in friction pressures.

53 If a fluid motor be introduced between the two sections where pressures are taken it will create a difference in pressure. This difference in pressure times the volume of fluid passing through the motor times the constants for heaviness of fluid, area, etc., gives the horsepower produced by the motor, or given to it if power is being taken from the fluid. But for the pressure caused by the motor, the sums of pressures at the two sections would be equal.

$$\frac{\text{cu. ft.}}{\text{const.}}\left[-\Big(v''+(-s''-f'')\Big)+(V''+S''+F'')\right]=\text{h.p.}$$

$$\frac{\text{cu. ft.}}{\text{const.}}\Big[V''+S''+F''+s''+f''-v''\Big]=\text{h.p.} \qquad [9]$$

$$\frac{\text{cu. ft.} \times \text{difference in pressure} \times 62.3}{33000 \qquad 12}=\text{air h.p.}$$

$$\frac{\text{cu. ft.} \times \text{difference in pressure}}{6356}=\text{air h.p.}$$

v'' = velocity, in. suction
s'' = static, in. of water
f'' = friction, in. from fan inlet to measurement on suction
V'' = velocity, in. water in discharge
S'' = static, in. water in discharge
F'' = friction, in. at discharge from point of measurement to fan (see Fig. 5)

When the friction heads F'' and f'' are considered as belonging to the air motor, then

$$\frac{\text{cu. ft.}}{\text{const.}}\Big[V''+S''+s''-v''\Big]=\text{air h.p.} \qquad [10]$$

54 When the discharge pipe and suction pipe are of the same diameter $V''-v''=0$ and equation [10] becomes

$$\frac{\text{cu. ft.}}{\text{const.}}\Big[S''+s''\Big]=\text{air h.p.} \qquad [11]$$

55 It is quite usual in buying and selling fans to understand that a certain effect on discharge shall be produced; that is, such a volume and pressure in discharge pipe.

56 When the work sold for is to be done on the discharge side the inevitable friction loss in drawing air through the inlets is not to be measured in. It is not wanted although as in the case of some single inlet fans it may be a very appreciable factor. Large inlets reduce it. For this case,

$$\frac{\text{cu. ft.}}{\text{const.}}\left[V''+S''+F''\right]=\text{air h.p. on discharge side}\ldots.[12]$$

and

$$\frac{\text{air h.p.}}{\text{h.p. to fan}}=\text{Efficiency for discharge side}$$

For a fan that does its work on the suction side with a short outlet on the discharge the same size as the inlet, a usual arrangement, we have

$$\frac{\text{cu. ft.}}{\text{const.}}\left[V+0+0+s''+f''-v''\right]=\text{air h.p. from [9], or when } V''=v''..[13]$$

$$\frac{\text{cu. ft.}}{\text{const.}}\left[s''+f''\right]=\text{air h.p. on suction side}\ldots\ldots\ldots[14]$$

If it is agreed to use an expanding or enlarged outlet such as the chimney of mine fans, equation [13] with V'' due to velocity at the outlet of the chimney should be used.

57 It might be agreed to sell for work on the suction side alone, then $\frac{\text{cu. ft.}}{\text{const.}}\left[s''+f''-v''\right]=$ air h.p. and it would be up to the fan maker to reduce V'' as much as possible.

58 When a fan is to do work on both sides the equations [9], [10] and [11] ought to be used to measure the air h.p.

59 A cone on outlet will not increase the efficiency of a fan but will decrease work to be done, that is, the same volume of air can be moved through the same resistance by less power, the speed being a little slower.

60 For air measuring work the formula $v=\sqrt{2gh}$ may be written

$$V=1097\sqrt{\frac{\text{vel.}''\text{ W. G.}}{\text{wt. cu. ft. air}}}\ldots\ldots\ldots\ldots\ldots[15]$$

When head due to velocity is expressed in its equivalent inches water gage readings, and V = Vel. in ft. per min.

$V = 4100 \sqrt{V''}$ W.G. for standard weight air of 0.0715 lb. per cu. ft.

AVERAGE VELOCITY OF FLOW IN A PIPE

61 The velocity is least at the surface of the pipe because of skin friction. In a square pipe the mean of the velocities and of the

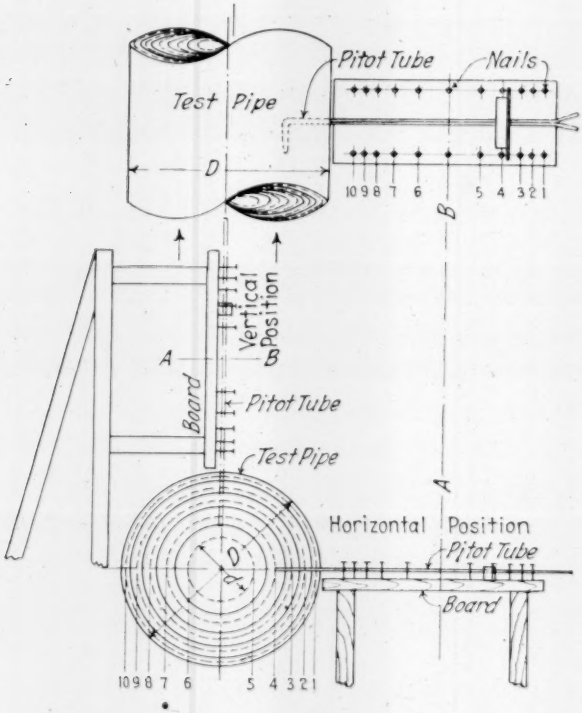


FIG. 12 SETTING PITOT TUBE

pressures taken at the center of a large number of equal squares into which the pipe is divided, is the average velocity and pressure in the pipe. In the case of a round pipe, divide into equal annular areas, and for each read at a point where the area inside equals the area outside (see Fig. 12). A convenient rule for obtaining diameters of circles is as follows:

62 Divide the diameter of the air pipe by the square root of twice the number of readings to be made from the circumference

to the center for the diameter of the smallest circle. Multiply this reading by $\sqrt{3}$, $\sqrt{5}$, etc., for diameters of larger circles. For example, for $48\frac{7}{16}$ diameter of pipe and eight readings $\frac{48\frac{7}{16}}{\sqrt{2\times 4}} = 17\frac{1}{8}\times \sqrt{3},\ \sqrt{5},$ etc. $= 29\frac{5}{8}$, $38\frac{1}{4}$, etc. This is done very readily on the slide rule by dividing the diameter pipe found on the lower scale by the square

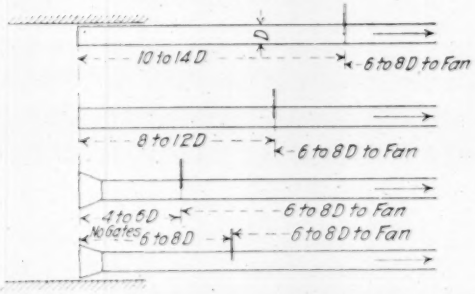


FIG. 13 LENGTH FOR TEST PIPE ON SUCTION

TEST IN DISCHARGE PIPE 20 DIAMETERS LONG. READINGS ⅝ IN. LENGTH FROM FAN

Position of pitot tube across pipe..	1	2	3	4	5	6	7	8	9	10
Velocity pressure in test pipe, in. .	0.730	0.885	0.990	1.025	1.060	1.075	1.055	1.022	0.925	0.680
Velocity, ft. per min.....	3503	3850	4080	4150	4222	4260	4220	4150	3950	3380

Mean velocity pressure in test pipe, 0.9447 in.; mean velocity, 3976.5 ft. per min.
$4100\times\sqrt{0.9447}$ in. = 3985 ft.

root of the numbers found on the slide. Without moving the slide multiply by $\sqrt{3}$, $\sqrt{5}$, etc., which roots are also found on the slide.

63 This rule may be demonstrated as follows: Divide the annular rings in two by a vertical line through the center. (See Fig. 12.)

$$\text{Total area} = A = \frac{\pi}{4}D^2$$

Circle d contains two of the equal areas of the 20 into which D is divided.

$$a = \text{area of small circle} = \frac{\pi d^2}{4}$$

$$\text{Unit area } \frac{a}{2} = \frac{A}{20}$$

$$A = \frac{a}{2} \times 20 = 10a,$$

$$10d^2 = D^2$$

since the area of a circle varies as the square of the diameter.

$$d^2 = \frac{D^2}{10}$$

$$d = \frac{D}{\sqrt{10}}$$

64 Since in equation [15] the velocity changes as the square root of the velocity water gage the velocities should be computed for each unit area and those averaged. The error is small, however, if the velocity be computed from the average of the velocity inches, ordinarily being less than 1 per cent from this cause.

65 There are usually irregularities of flow in air entering a pipe and if it is the discharge from a fan the irregularities may be considerable; if an inlet pipe there will be less and if fitted with cone inlet there will be considerably less irregularity of flow. This irregularity becomes less away from the pipe inlet and in 8 to 12 diameters in the case of fan discharge and in straight pipe good pitot tube readings may be taken. (See Fig. 13.)

66 For accurate measurement it is recommended that both vertical and horizontal readings across the pipe be taken rather than either alone, especially if there has been a bend in the pipe or if the pipe is the discharge from a fan.

APPENDIX

CALIBRATION OF STATIC AND IMPACT ORIFICES

67 This calibration was made by revolving the various tips at the end of a slender arm and noting the speed and the pressures at the axis. The arm was made of brass tubing $\frac{3}{16}$ in. and $\frac{3}{8}$ in. in diameter, sweated together. Stay wires were used to give it rigidity, as shown in Fig. 2, and the tubing was fastened to its countershaft by the set screw of the collar through which the larger tube fastens. The different tips tried were soldered to the end of this arm. A very small air-tight connection or stuffing box at the axis was afforded by a $\frac{1}{8}$-in. brass tube projecting from the larger tube along the axis and having a ring of greased candle wicking on it against which the cone end of a larger tube was pressed, the small tube revolving in it. The arm was counterweighted at W. Care was taken to press on the outer tube only hard enough to keep the connection tight.

68 The revolutions were counted by a hand-speed counter at the other end of the shaft, usually for two or three minutes. The length of arm was measured from axis to center of orifice by steel scales. The shaft was driven by an automatic engine having no other load so that the speed was very steady and uniform.

69 Pressure was taken by a vertical water manometer of large tube area (0.45 in. inside diameter), in most of the runs. An inclined gasolene manometer was used, in those of small head the reading being corrected to the head of water. Temperature and humidity were taken in the room not far from the arm.

70 The $\frac{1}{4}$-in. tubes at the end of the arm which were used in all the runs except the last four projected forward from the arm about $3\frac{1}{4}$ in. and backward about $2\frac{1}{2}$ in. The static holes in the sides were about 2 in. ahead of the arm. Different tubes were used but were adjusted alike, being soldered to the corner braces and pipe.

71 Air being elastic, that portion nearer to the shaft presses, due to centrifugal force, on the air further out in the tube compressing it slightly. Therefore the outer portion of the air in the tube is slightly more dense than the inner, and this varying density makes it necessary to go to calculus for a solution.

72 For an inelastic fluid the computation would be very much simpler, and for a rotating column of unit area it would be as follows:

$$\text{Centrifugal force (pressure per sq. ft.)} = \frac{wv^2}{gr} = \frac{w(\pi 2rn)^2}{gr} = \frac{2ry(\pi 2rn)^2}{gr}$$

w = total weight of fluid in lb.
v = velocity in ft. per sec.
g = gravity
r = radius to center of gravity
y = weight per cu. ft. of fluid
n = revolutions per second

DIFFERENCE IN PRESSURE BETWEEN THE ENDS OF A REVOLVING RADIUS OF ELASTIC FLUID

73 Assume
PV = constant
p = pressure in lb. per sq. ft. at any intermediate point, variable
P_o = pressure in lb. per sq. ft. at axis due to the atmospheric pressure
P_t = pressure in lb. per sq. ft. at the free end of the radius—at tip
W = weight per cu. ft. of elastic fluid, variable
W_o = weight per cu. ft. of elastic fluid at atmospheric pressure
N = r.p.m.
R = radius in ft.
R_o minimum radius
R_t = maximum radius
$B\alpha$ = barometer at time of experiment

74 For a very short radius of elastic fluid

$$p = \frac{WR\left(2\pi\frac{RN}{2\ 60}\right)^2}{g \times \frac{R}{2}} = \frac{2\pi^2 WR^2N^2}{g \times 3600}$$

$$dp = \frac{2 \times 2\pi^2 WN^2}{g \times 3600} RdR$$

and as

$$\frac{W}{p} = \frac{W_o}{62.3 \times \frac{B\alpha \times 13.61}{12}}$$

$$dp = \frac{4\pi^2 \times 12 W_o N^2 pRdR}{g \times 3600 \times 62.3 \times 13.61 \times B\alpha} = C \times pRdR$$

$$\frac{dp}{p} = CRdR$$

$$\int_{P_o}^{P_t} \frac{dp}{p} = C \int_{R_o}^{R_t} RdR$$

inte grating

$$\log_e \frac{P_t}{P_o} = \frac{C}{2}(R_t^2 - R_o^2) + K$$

when $R_t = R_o$, $P_t = P_o$ and $\frac{P_t}{P_o} = 1$. Log of 1 = o and for this $(R_t^2 - R_o^2) = o$, K must = o. As $\log_e$ = common log × 2.302585 we have

$$\text{common log } \frac{P_t}{P_o} = \frac{C}{2 \times 2.302585}(R_t^2 - R_o^2) \quad \ldots\ldots\ldots\ldots \quad [18]$$

$\frac{P_t}{P_o}$ is the ratio of pressure at the ends of radius and may be expressed in any units of pressure or head.

$$\frac{C}{2 \times 2.302585} = \frac{2 \times \pi^2 \times 12 \times W_o \times N^2}{2.302585 \times g \times 3600 \times 62.3 \times 13.61 \times B\alpha} = 0.0000010479 \frac{W_o \times N^2}{B\alpha}$$

	No.	log	No.	log
	2	0.301030	2.302585	0.362210
	π^2	0.4971498	32.16	1.507316
		0.4971598	3600.	3.556303
	12	1.079181	62.3	1.794488
		2.3745106	13.61	1.133858
		8.354175		8.354175
0.0000010479		6.0203356		

Equation [18] with numerical constants is

$$\text{common log } \frac{P_t}{P_o} = \log \frac{\text{in. mercury}}{B\alpha} = 0.0000010479 \frac{W_o N^2 R_t^2}{B\alpha}$$

75 *Impact Orifice.* When the pressure at the axis is atmospheric, or zero on the manometer then the common log of pressure at the tip in inches of mercury is

$$0.0000010479 \frac{W_o N^2 R_t^2}{B\alpha} + \log B\alpha \quad \ldots\ldots\ldots\ldots\ldots \quad [19]$$

The number corresponding to log $B\alpha$ + the numerical factor, the second part of equation [19] less the barometer pressure is the difference in pressure between the two ends, or the centrifugal head.

76 *Static Orifice.* If a static orifice is correct the pressure inside the tip = pressure outside; or atmospheric, at all velocities of air past; then suction k on the manometer at the axis = centrifugal head in the arm. For this case

$$\log \text{ pressure at tip} = 0.0000010479 \frac{W_o N^2 R_t^2}{B\alpha} + \log (B\alpha - K) \ldots\ldots \quad [20]$$

and the number corresponding to the second part of equation [20] less the pressure at the axis $(B\alpha - K)$ is the centrifugal head, K and $B\alpha$ of course being expressed in the same unit of pressure as mercury.

77 *Impact.* Of the runs for impact the first nine were with a fan drawing away the disturbed air. Of these the ones with full gate were affected only very slightly, if at all, by the air motion due to the rotating arm. This point was determined by testing with smoke. The air travel toward the inlet was probably never anywhere more than 300 ft. per min. At the smaller gates there was, however, some motion of the air with the arm so that the runs with full gate deserve more consideration that those at small gate.

TABLE 1 PITOT TUBE CALIBRATION

Test No.	R.P.M.	*A* Manometer Reading, In. Water	*B* Centrifugal Head, Computed	*A*+*B* Pressure in Tip Back of Orifice	*C* Pressure Due to Velocity of Tip in Still Air, In. Water	Pressure in Tip ÷ Pressure Due to Velocity *A*+*B*÷*C*	Weight of 1 Cu. Ft. of Air	Humidity Temperatures Dry	Humidity Temperatures Wet	Barometer	Length of Arm Axis to Tip, In.	Date (1908) and Time	Remarks
1	129¾	00	0.489	0.489	0.4888	1.000	0.07247	75°	60°	29.34	41.656	10/15	Impact orifice; ½ gate on fan
2	328½	0	3.112	3.112	3.095	1.003	0.07247	75°	60°	29.34	41.656	a.m.	Impact orifice; full gate
3	388	0	4.4	4.4	4.371	1.004	0.07247	75°	60°	29.34	41.656	a.m.	Impact orifice
4	0 to 480	0	.....	.....	.....	.....	0.07247	75°	60°	29.34	41.656	a.m.	Impact orifice; slow up to full speed. Air motion slow and nearly radial to shaft
5	361	a little +	3.727	3.727	3.709	1.005	0.07192	79°	64°	29.36	41.656	2 p.m.	Impact orifice; air currents change slightly about inlet
6	482	a little −	6.665	6.665	6.612	1.008	0.07192	79°	64°	29.36	41.656	.....	Impact orifice
7	430¾	$-\frac{1}{8}$ to $-\frac{7}{32}$	5.31	5.18	5.27	0.984	0.07182	79°	64°	29.32	41.656	10/14	Impact orifice, but ½ gate
8	414	0	4.905	4.905	4.870	1.006	0.07182	75°	64°	29.32	41.656	10/14	Impact orifice, but full gate
9	414	$-\frac{1}{8}$	4.905	4.78	4.870	0.983	0.07182	75°	64°	29.32	41.656	10/14	Impact orifice, but ¼ gate
10	464	$-\frac{3}{16}$	.....	.....	.....	.....	0.06997	92°	78°	29.40	41.62	7/6	$\frac{1}{32}$ in. impact orifice; Std.
11	464	$-\frac{7}{32}$	.....	.....	.....	.....	0.06997	92°	78°	29.40	41.62	p.m.	5/16 in. diameter; flat point
12	506.5	−0.23	.....	.....	.....	.....	0.07152	82°	69½°	29.43	41.7	8/15	¼ in. tube open
13	505.5	−0.24	.....	.....	.....	.....	0.07177	78½°	67½°	29.33	41.7	8/18	Std. impact orifice
14	504	$-8\frac{1}{4}$	7.185	−1.065	÷ *B*	−0.148	0.07202	78°	67°	29.41	41.80	8/26	Back end of tip open; ¼ in. tubing tip square
15	506	$-9\frac{3}{8}$	7.21	−2.165	÷ *B*	−0.305	0.07202	78°	67°	29.41	41.80	3.50 p.m.	Same; but with cord around tube at tip $\frac{11}{32}$ in.

16	508	−8.515	7.28	−1.235	÷ *B*	−0.1695	0.07202	78°	67°	29.41	41.80	.....	5/16 in. tube square as was ¼ in.
17	503.5	−8.44	7.19	−1.25	÷ *B*	−0.174	0.07202	78°	67°	29.41	41.80	.....	¼ in., tube, but beveled 45 deg.
18	507	−0.328	7.4	7.072	7.02	−1.008	0.07202	78°	67°	29.41	41.67	4.00 p.m.	Std. impact; air travel with arm; 245 ft. by anemometer
19	142	−0.562	0.572	0.010	÷ *B*	+0.0175	0.07199	77½°	65½°	29.34	41.625	7/6	Std. static—holes 0.02 in. in diameter
20	193.5	−1.047	1.062	0.015	1.065	+0.014	0.07199	77½°	65½°	29.34	41.625	p.m.	Std. static—holes 0.02 in. in diameter
21	242.2	−1.6875	1.667	−0.020	÷ *B*	−0.012	0.07199	77½°	65½°	29.34	41.625	p.m.	Std. static—holes 0.02 in. in diameter
22	508	−7.32	7.27	−0.05	÷ *B*	−0.0069	0.07208	76°	64¼°	29.29	41.62	8/20	Std. static—holes 0.02 in. in diameter
23	511	−7.34	7.375	+0.035	÷ *B*	+0.0048	0.07208	76°	64¼°	29.29	41.62	5.15 p.m.	Std. static—holes 0.02 in. in diameter
24	512	−7.36	7.4	+0.04	÷ *B*	+0.0054	0.07208	76°	64¼°	29.29	41.62	.....	Std. static—holes 0.02 in. in diameter
25	513	−7.405	7.385	−0.02	÷ *B*	−0.0028	0.07208	76°	64¼°	29.29	41.62	.....	Std. static—holes 0.02 in. in diameter
26	515	−7.46	7.48	+0.02	÷ *B*	+0.0028	0.07208	76°	64¼°	29.29	41.62	.....	Std. static—holes 0.02 in. in diameter
27	505.5	−7.125	7.23	+0.105	÷ *B*	+0.014	0.07216	77°	67°	29.41	41.67	8/26	Std. static—2 holes 0.026 in. in diameter and 2 −0.02 in. and
28	507.5	−7.156	7.28+	+0.124	÷ *B*	+0.017	0.07216	77°	67°	29.41	41.67	2 p.m.	Std. static inner tube—⅛ in. in diameter
29	507.3	−7⅛	7.28	+0.092	÷ *B*	+0.0126	0.07216	77°	67°	29.41	41.67	.....	Std. static inner tube used in Nos. 27 to 33
30	504.6	−7.125	7.2	+0.075	÷ *B*	+0.010	0.07202	78°	67½°	29.41	41.67	.....	2 holes 0.02 in. in diameter and inner tube ⅛ in. in diameter
31	506.5	−7.156	7.255	+0.099	÷ *B*	+0.0136	0.07202	78°	67½°	29.41	41.67	.....	
32	504	−7.156	7.19	+0.034	÷ *B*	+0.0047	0.07202	78°	67½°	29.41	41.67	.....	
33	506	−7.187	7.25	+0.063	÷ *B*	+0.0087	0.07202	78°	67½°	29.41	41.67	3 p.m.	Same as last; very small leak stopped
34	463	−5.396	6.035	+0.639	÷ *B*	+0.106	0.07050	86°	70°	29.20	41.76	7/10	⅛ in. pipe about 0.28 in. diameter inside with 2 slots opposite 1/16 in. x 2½ in. —static orifice—mean of 4 tests
35	511.2	−7.23	7.33	+1.05	÷ *B*	+0.0143	0.07090	81°	75°	29.21	41.76	8/13, a.m.	Above, with ¼ in. tube inside central
36	501.5	−6.65	7.08	+0.43	÷ *B*	+0.061	0.07057	84½°	75°	29.24	41.76	8/13, 3 p.m.	Above, with inner tube 0.02 in. smaller
37	502.5	−6.4	7.10	+0.70	÷ *B*	+0.099	0.07057	84½°	75°	29.24	41.76	8/13, 3.30 p.m.	Above. No inner tube—same as No. 34

No leak at axis. No inner tube.

78 Another minor source of trouble was the tendency of the stuffing box to heat the air in the tube slightly. This was largely overcome by the very small size and by care to use soft grease freely on the wicking, also by not pressing the outer tube against the wicking harder than necessary. Between runs outside air was allowed to replace the warmed air, but doubtless the results are affected very slightly by the air in the tube being a little warmer than that in the room. The brass tube of the arm always felt as cool after a run as before. Warming of the brass tubes of the stuffing box to as slight an extent as 5 is noticeable to touch at these temperatures.

79 An increase of 10 deg. in temperature in run No. 8 without change of moisture in the air in arm, changes the computed centrifugal head from 4.905 in. at room temperature to 4.825 in.

80 Of the first nine runs only No. 6 at the highest speed shows a negative pressure in the water manometer used, when the fan was fully open. With the smaller gates the readings were slightly negative. The mean of these runs is 0.9991 for the pressure in the tip divided by the pressure due to velocity. However, considerably more weight must be given the runs with full gate.

If the runs with the smaller gates be left out then the pressure in the tip exceeds that due to velocity by an amount that would have been noticed and measured, for the water levels in the manometer used were steady and the conditions good. It is believed that the slight rise in temperature of the air in the tubes accounts for the observed pressure being slightly less than the computed.

81 Runs Nos. 10 to 18 inclusive were all made without the fan to draw the disturbed air away. Nos. 10 and 11 show a comparison of a flat impact point $\frac{5}{16}$ in. in diameter with a hole in the center about $\frac{1}{16}$ in. in diameter with the same tip used in the first runs, marked Std. Nos. 12 and 13 compare a $\frac{1}{4}$-in. open brass tube with end filled square with Std.

82 Nos. 14 to 17 were with the back end of the tip open (see *B* in Fig. 2), using four forms of opening. No. 18 is given to show about what air motion the arm set up for these runs. These readings are given because nearly every one interested in the pitot tube sooner or later wants to know what this suction is. The suction evidently depends for one thing on the shape of the orifice and the tip.

83 In No. 18 the pressure due to velocity may be found from the velocity of tip less the air motion

$$\text{Tip velocity} = \frac{41.67 \text{ in.} \times \pi\, 2 \times 507}{12} = 11080$$

$$\text{Air motion in path and moving with tip} = 245$$

$$\text{Velocity of impingement on tip} = 10835$$

From

$$\text{velocity, ft.} = 1097 \sqrt{\frac{\text{velocity pressure}}{\text{weight 1 cu. ft. air}}}$$

$$\left(\frac{10835}{4085}\right)^2 = 7.02 \text{ in.}$$

which is the velocity pressure due to the relative velocity of the tip to the air in inches of water. The centrifugal heads in column *B* were computed by logarithms by the method in Par. 74 except that the final head in inches of mercury was multiplied by 13.61 by slide rule.

84 For computation of centrifugal head for the static runs pressure at the axis is $B\alpha - k$, if the pressure in the tip is atmospheric, and the mean density in the arm is slightly less than for the impact runs, since for the impact runs pressure at the axis is very nearly atmospheric.

85 It can be shown that the pressure due to impact on the tip, Fig. 2, would exactly equal the centrifugal head if the fluid in the arm were of uniform density.

86 *Static.* Static runs Nos. 19 to 33 are with good forms of static orifice. It is thought that the two sets of runs 22 to 26 and 32 and 33 deserve to carry greater weight than the others. However, the mean of all for pressure in tip divided by the centrifugal head is 0.00625. It is evident by comparing the first part of these runs with the last that the holes are small enough so that the static does not depend on the inner tube in any way.

87 The static orifices are simply holes 0.02 in. in diameter drilled through the tubing which was about 0.03 in. thick. The holes were clean and the outer edges square, the little burr being carefully filed off.

88 Runs Nos. 34 to 37 are with slotted static orifice. The outer tube was about 0.4. in. in diameter, its inside diameter being about 0.28 in. With no inner tube the results are poor even with this thick tube. With an inner tube central and fitting the outer within about 0.015 in. the result was fairly good, but when the inner tube was made 0.02 in. smaller in diameter the result was poor again. It is very interesting to know that when fine mesh wire cloth was wrapped around the tube outside the slots the results are good and agree with those from the small drilled holes. Just how accurate the gauze static would be was not found.

89 By comparison with standard static in air flowing in a pipe the results are found to be considerably worse with a thin outer tube than with a thick one using the same inner tube. An instrument with $2\frac{1}{4}$-in. slots $\frac{1}{16}$ in. wide in $\frac{3}{8}$ in. tubing $\frac{1}{32}$ in. thick and with $\frac{7}{32}$ in. inner tube gave static pressure readings 14 per cent of the velocity pressure too high.

90 By comparison in this way it was also found that holes $\frac{1}{16}$ in. in diameter in the $\frac{1}{4}$ in. regular tip which is $\frac{1}{32}$ in. thick give poor results.

THE PRESENT STATE OF DEVELOPMENT OF LARGE STEAM TURBINES

By A. G. Christie, Published in The Journal for May

ABSTRACT OF PAPER

This paper deals with the present state of development of leading types of large steam turbines, some details of construction, the commercial results obtained and some new uses to which steam turbines have been put. It also points out the probable tendencies of steam-turbine development.

For the purposes of discussion, large steam turbines are divided into two types, fundamental and modified or combined. The weak elements of the fundamental types are discussed and the advantages of the new types pointed out. Some new constructions are shown in section.

The details of construction are discussed fully. A table with brief notes on the details of construction of individual types is included for rapid comparison of the practice of various manufacturers. The different types are also compared on the basis of efficiency as shown by published results of tests.

The present status of low-pressure turbines, turbo-compressors, turbo-driven pumps, geared turbines and marine turbines is discussed briefly. The closing paragraphs deal with the probable trend of steam turbine development and future possibilities.

CLOSURE[1]

The Author. In the discussion it has been held that as high an efficiency ratio cannot be obtained with high vacuum and low superheat as with moderate vacuum and high superheat. This is probably true of the Curtis turbine, since it does not apparently utilize high vacuum as economically as other types, which was pointed out in the paper itself. This is therefore a characteristic only of this type of turbine.

If a turbine is designed for a large amount of available energy there is no reason why fully as high an efficiency ratio should not

[1] The discussion was published in The Journal for August.

Presented at the Spring Meeting, Cleveland, 1912, of The American Society of Mechanical Engineers, 29 West 39th Street, New York. All discussion is subject to revision.

be obtained as with a small amount of available energy. Sufficient data is not at hand to warrant a general statement for all types that such conditions as above referred to cannot be fulfilled.

In regard to the increasing use of the efficiency ratio as a measure of steam turbine economy, it is interesting to note that a leading European firm includes a statement in their steam turbine specifications to the effect that should the steam or vacuum conditions at the time of the tests vary from those specified, the efficiency ratio of the unit shall not exceed that represented by the guarantee.

The author recognizes that the efficiency ratio alone does not provide all the information desirable for a given turbine, but when used in conjunction with specified steam and vacuum conditions, it provides the most reliable index of the actual performance of the steam turbine.

It is very gratifying to note the attention given to the present uncertain state of steam turbine ratings. It is to be hoped that the Committee on Power Tests of the Society will bring in an early report on this subject and fix certain standards of rating.

Objections have been raised to the Erste Brünner tests. Before including these in Table 2, inquiry was made as to their reliability. These tests were conducted by Prof. K. Koerner, one of the most prominent of Austrian engineers, who substantiated the figures quoted.

Mr. H. Zoelly presented a paper[1] at the 1911 meeting in Zurich, Switzerland of the Institution of Mechanical Engineers in which he made the statement quoted in Par. 51. He gave his reasons for such an opinion and also presented results to verify his statements. An abstract of this paper was printed in Engineering, April 29, 1911, and can be referred to by those interested.

Mr. Crissey presents a very excellent classification of steam turbines. However, for the purpose of this paper, the author preferred to classify turbines not so much as pure theoretical types, but under a classification of names and types with which engineers in general are familiar, hoping thereby to avoid confusion.

There seems to be a diversity of opinion regarding blade erosion in Curtis turbines. Some users have reported serious

[1] Steam-Turbines, Proc. I. Mech. Engrs., 1911, No. 3, p. 681.

erosion in much less time than Mr. Moyer quotes. These difficulties are being steadily overcome by improvements in design.

Regarding Mr. Hodgkinson's statement concerning blade troubles, the author fails to recall in his own experience, under ordinary conditions of operation, a case of blade failure due to centrifugal force alone.

The value of test results supplied by manufacturers has been criticised. There have been cases in the past where such results have been questionable, but in general manufacturers now find it advisable to publish only such tests as represent the actual and usual performance of their machines. It is frequently the case that steam turbines sold subsequent to the publication of such tests must operate with the same degree of economy as represented by the test results.

The corrections for Table 1 given by Mr. Hodgkinson are very welcome, bringing the data presented strictly up to date. The description of the new type of Westinghouse blading is also of considerable interest to engineers.

In connection with Dr. Stodola's tests of a Sulzer turbine quoted by Mr. Moyer, it was the author's impression when visiting Sulzer's works that while their Diesel engines probably excelled all other makes, their steam turbines were not representative of the best advanced practice in Curtis-Parsons construction, owing to characteristics of general design and particularly to their Parsons blading. This blading was much heavier in section than the usual form, and was not shrouded or otherwise reinforced. The tests quoted amply bear out this impression, and the author believes that such results should not be considered as typically representative of the whole class of Curtis-Parsons turbines.

The test results presented by Mr. Emmet for Curtis turbines show decided improvements in design over the earlier Curtis turbines in Table 2, but still do not exceed the Curtis-Parsons or Curtis-Rateau results. Table 4 presents some results of tests which were available subsequent to the preparation of Table 2, and which show some remarkable performances of Zoelly turbines. The Bellis and Morcom tests were on some of their earlier machines, and it is reasonable to expect considerable improvement in units now under construction. The results obtained by the Ljungström turbines are remarkable for high efficiency.

TABLE 4 ECONOMY TESTS OF HIGH PRESSURE STEAM TURBINES

EFFICIENCY RATIOS BASED ON E. H. P.

Maker of Turbine	Type	Date of Test	Load, Kw.	R.P.M.	Steam Pressure, Lb. Absolute	Temperature at Throttle, Deg. Fahr.	Vacuum Referred to 29.92 In. Barometer	Condenser Pressure, Lb. Absolute	Lb. of Steam per Kw-Hr.	B.t.u. per Kw-Hr.	Heat Utilized per Lb. of Steam	Heat Available per Lb. of Steam	Efficiency Ratio	Reference
Escher, Wyss & Co.	Zoelly	1912	1275	3000	196.3	588	28.99	0.455	12.59	15940	268.6	418.1	64.3	Zeit. d. v. d. Ing., 4 13 '12
Escher, Wyss & Co.	Zoelly	1912	5418	1500	180.7	588	28.74	0.580	11.95	15060	285.5	402.1	71.0	
Elsassische Maschinenbau Gesellschaft	Zoelly	1912	10006	1250	167.8	565	28.65	0.625	12.01	14975	284.2	389.1	73.0	
Belliss & Morcom	Drum Impulse	1911	3000	1500	192.5	558	27.92	0.981	15.0	18420	227.3	372.7	61.0	Manufacturers' Data
Belliss & Morcom	Drum Impulse	1911	2500	1500	164.7	466	26.92	1.471	16.95	19815	201.2	323.6	62.2	
Aktiebolaget Ljungström Angturbin...	Radial Reaction (Ljungström)	1911	1000	3000	181.7	669	28.55	0.671	11.68	15130	292.0	414.9	70.4	Engineering, 4 19 '12
		1911	972	3000	176.7	667	28.58	0.657	11.55	14960	295.3	414.2	71.3	

Marks and Davis tables used.

The design of the details of this turbine shows a high degree of originality on the part of the inventor. The turbine is said to be very cheap to build, but before putting it on the American market many parts would have to be simplified and improved, particularly the bearings, steam packing and blade holding rings.

BITUMINOUS COAL PRODUCERS FOR POWER

By C. M. Garland, Published in The Journal for June

ABSTRACT OF PAPER

The paper describes the apparatus and general arrangement of bituminous coal producers as designed for power. The scrubbing apparatus is described in detail, data given on its efficiency, and the amount of solid matter delivered in the gas. Data on the efficiency of the plant, composition of the gas, and operating costs, together with brief discussions on these items are also included. Figures for the first cost and operating costs at full load for a 1200-b.h.p. plant are given in such a way as to make them applicable to different conditions of fuel and load.

CLOSURE [1]

The Author. Regarding the question of the low grade fuels high in ash, brought up by Mr. Hood, the writer would state that in this country at the present time there are producers operating on fuels containing as high as 22 to 23 per cent of ash. These are principally anthracite fuels and the producers that have proved successful in using these fuels are of the rotating table type, in which an ash bar rakes the ash from the table, dropping it into a water seal or a cone bottom. It is quite probable that this same type of producer will use fuels successfully that contain ash up to 30 and 40 per cent.

The CH_4 as given in the paper in the different analyses, was carefully determined and the writer feels that the results are accurate for two reasons. First, the calorific value of the gas, as calculated from the analyses, checked within about 1 per cent of the calorific value of the fuel as carefully determined by a Junkers calorimeter. This would indicate that no serious error was made in the determination of the CH_4. The carbon monoxide

[1] The discussion was published in The Journal for August.

Presented at the Spring Meeting, Cleveland, 1912, of The American Society of Mechanical Engineers, 29 West 39th Street, New York. All discussion is subject to revision.

was absorbed in three successive pipettes, so that there is every reason to believe that essentially the last trace of this constituent was removed. The writer has conducted a number of investigations on the accuracy of the removal of the carbon monoxide from producer gas and finds that if reasonable care is exercised in the analysis there is no danger whatever of having this constituent appear in the determination of the CH_4. To insure this, however, three pipettes must be used and the solution in the third pipette maintained fresh.

Referring to Mr. Marsh's remarks regarding the question of the use of producers for peak loads of 10,000 kw. and over, and also to Mr. Dreyfus' remarks along the same lines, the writer can only reply that under certain conditions a producer plant is economical and desirable for such loads. Under ordinary conditions, however, the large central station in cities is operating under a very low load factor, probably under 50 per cent, and due to this low load factor the straight producer plant cannot compete with steam turbines. Where the load factor, however, is 75 per cent or over, the producer plant in practically every case would be more economical than a steam plant. For plants of 1000 kw., it can be said at the present time there are a number of such installations in operation that are producing highly satisfactory results. For plants of this size and with a load factor not under 50 per cent the producer plant will show a decided saving over a steam plant. The writer feels that Mr. Dreyfus is in error when he states that with the prevailing price of fuel a producer is commercially limited to sizes under 1000 kw.

Mr. Baker's statement that the temperature of 212 deg. is sufficient to prevent corrosion in exhaust fire boilers is correct, providing the load on the engine is uniform so that in the case of load fluctuation the temperature of the gas will not be decreased below this point. In actual practice, however, there is always more or less fluctuation in the load, so that the heating surface in the boiler should be so designed that the minimum temperature should be 212 deg. This will, of course, mean that the maximum temperature of the gas leaving the boiler will be much higher than this, anywhere from 220 to 300 deg. fahr. It is, therefore, this practical consideration that induces the writer to state that the gas should not be reduced below 220 deg. fahr. The writer feels that the piston cited by Mr. Baker is a very

unusual case, for it would hardly be possible to honeycomb a block of steel in diluted sulphuric acid within 24 hours time, unless there were very unusual conditions of composition existing in the steel. The fact that the steel was honeycombed would seem to indicate that the steel was not of uniform composition. The acidity caused from the presence of sulphur in exhaust gases, so far as the writer's experience extends, does not produce serious corrosion on cast-iron parts.

FOREIGN REVIEW

BRIEF ABSTRACTS OF CURRENT ARTICLES IN FOREIGN PERIODICALS

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FOREIGN REVIEW

The aim of the Foreign Review is to present, within the available space, the main data contained in the articles indexed. Where possible, reference is made to English or American publications containing fuller information on the subject treated. Measures are given both in original units and their English equivalents. In many instances, engravings and tables are reproduced. Opinions expressed are those of the reviewer, not of the Society. Articles are classified as *c* comparative; *d* descriptive; *e* experimental; *g* general; *h* historical; *m* mathematical; *p* practical; *t* theoretical. Articles of exceptional merit are rated *A* by the reviewer.

Aeronautics

COMPARISON OF EXPERIMENTS WITH AN AIRSHIP AND MEASUREMENTS MADE ON ITS MODEL (*Vergleich von Versuchen auf einem Flugzeug mit Messungen an dessen Modell*, C. Wieselsberger. *Zeits. für Flugtechnik und Motorluftschiffahrt*, vol. 3, no. 13, p. 174, July 13, 1912. 2 pp. 3 figs. *e*). Account of a paper published by G. Eiffel in the "*Inauguration du nouveau laboratoire aérodynamique*" on the comparison of data obtained from actual measurement on an airship specially fitted with proper measuring appliances and data obtained in the laboratory on a model of the same airship placed in an artificial air current. This question is very important at the present time in view of the installation of numerous aerodynamic laboratories in various parts of the world. It was found that the values obtained in the laboratory are surprisingly close to those obtained in actual flights, the two, on the average of seven measurements, being within 1 to 2 per cent of each other, which is close enough for all practical purposes, and proves that laws of flight may be profitably investigated on models in laboratories. According to the law of similarity, a model tested at the same velocity as an airship of working size could not be expected to give absolutely the same results, since this is possible only if the respective velocities are inversely proportional to the length of the machines, which cannot always be done conveniently because with high velocities in actual flight and models of moderate size the velocities of air in the laboratory work might approach to that of sound, and then, according to Blasius, the elasticity of air could no longer be neglected.

The article contains also a description of a simple application of the

Venturi tube for measuring the velocity of an airship with respect to the air. The air flows through a nozzle (Fig. 1) consisting of two truncated cones, with a U-shaped manometer tube attached at *A* and *B* as shown. The pressure difference *p* is proportional to the square of the velocity of the airship with respect to the air. The constant of the instrument is determined by calibration. The position of the meniscus may be recorded by photography.

Air Machinery

FREE AIR OR CIRCULATING AIR? (*Frischluft oder Zirkulationsluft?* Oscar Gerold. *Dinglers polytechnisches Journal*, vol. 327, no. 29-31, July 20-August 3, 1912, 7 pp. *p*). Discussion of *ventilating problems in textile plants*. Exhaust fans in textile plants have the disadvantage of carrying away not only air containing dust and lint, but also the so-called false air, i. e. comparatively clean air, the handling of which involves unnecessary loss of power, besides other disadvantages. A better system there-

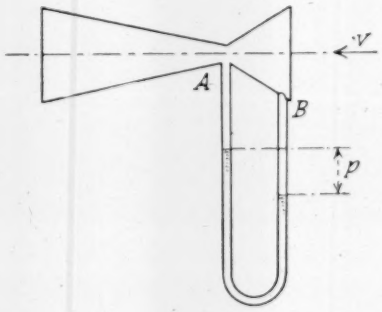

FIG. 1 VENTURI TUBE FOR MEASURING THE VELOCITY OF AN AIRSHIP WITH RESPECT TO THE AIR

fore appeared to be to place suction apparatus at the dust-producing machinery so as to keep the lint out of the work rooms. The air carried away by these apparatus was however too dirty to be let free. It could be cleaned either in rotary strainers or in filters, the former delivering air if not absolutely pure, at least clean enough to be used without too bad consequences, while the filters delivered practically pure air. The exhaust fans of the suction apparatus carry however much more air than is really necessary to convey the dust, and while for hygienic reasons change of air three times per hour is quite sufficient, the action of the exhaust apparatus leads to the air being changed in the room sometimes as often as twenty times per hour. As a result immense amounts of properly conditioned, i. e. humidified and heated air are uselessly blown into the atmosphere, and have to be replaced by new conditioned air at a certain expense. When filters are used, the air is so pure that it may be returned to the workshop, with an addition of about 15 per cent new air, this amount being sufficient to provide for the three times per hour renovation of the air. The question as to which system to use, free air with rotary strainer (*Zyk-*

lonen) or air of circulation with filters, is said to be a burning one in Germany, the main argument of the manufacturers of free air apparatus being that air returned to the workshop from the filters is polluted by dust and carbon dioxide. The investigation of Dr. Götting, court chemist in Breslau, Germany, has shown however that filtered air does not practically differ from fresh air with respect to the contents of dust and carbon dioxide. The author proceeds therefore to compare the cost of installation and operation of both systems in Germany, and shows that with free air, as compared with air of circulation systems, the initial cost of this installation is about 40 per cent, and the cost of operation over 50 per cent higher.

Firing and Fuels

NEW METHOD OF FIRING WITH LIGNITE AND LIGNITE BRIQUETTES (*Ein neues Verfahren zur Verfeuerung von Braunkohlen und Braunkohlenbriketts*, Pradel. *Braunkohle*, vol. 9, no. 14, p. 209, July 5, 1912. 1½ pp., 2 figs. *d*). Description of a new furnace arrangement patented by Ernst Lochner, of Jena, Germany, for firing with lignite and lignite briquettes. It is a self-feeding furnace but so arranged that the front part of the grate is fully exposed to view, while the clinker grate is set in the coldest part of the furnace. To make this possible, the grate is inclined not from the front part towards the bridge as usual, but in the opposite way, so that its highest part which receives the fuel is located just under the fire bridge, and its lowest part, the clinker grate, at the fire door. These furnaces have been in actual service for several months, and are said to work satisfactorily, and to be especially good on the arched parts of the furnace.

Hydraulics

FUNDAMENTAL THEOREMS OF HYDRAULICS OF RIVERS (*Théorèmes fondamentaux d'hydraulique fluviale*, A. Merten. *Annales de l'Association des ingénieurs sortis des écoles spéciales de Gand*, vol. 5, no. 2, 1912, p. 207. 5 pp., 1 fig. *t*). The theorems of hydraulics relating to the flow of water in open channels are derived from an investigation of the flow of only prismatic bodies of water (canals), and are applied to the whole field by the use of a postulate claiming that it is legitimate to apply to the *flow of natural bodies of water* theorems deduced for prismatic bodies of water. The author claims to have formulated a theory by which the modifications produced in the axes by the variations of the volumes of flow may be directly considered. The differential equation of the axes is:

$$ids - dh\sqrt{1-i^2} = \frac{q^2}{g}\cdot\frac{d\omega}{\omega^3} + \frac{q^2}{\omega^3}bxds$$

or

$$i - \frac{dh}{ds}\sqrt{1-i^2} = \frac{q^2}{\omega^3}\left[bx - \frac{1}{g}\left(\frac{\delta\omega}{\delta s} + \frac{\delta\omega}{\delta h}\cdot\frac{dh}{ds}\right)\right]$$

or

$$i - \frac{dh}{ds}\sqrt{1-i^2} = \frac{q^2}{\omega^3}\left[bx - \frac{1}{g}\left(\frac{\delta\omega}{\delta s} + l\frac{dh}{ds}\right)\right]$$

where i is the slope, s the distance along the axis from some origin, h the elevation of the water surface, q volume of flow, ω wetted area, x wetted perimeter, b coefficient of resistance. The lines in Fig. 2 are drawn with $\frac{dh}{ds}$ as abscissae, and the values of the first and second members as ordinates. Line I (first member) the point of intersection with the axis of the abscissae, is independent of the volume of flow, elevation of water surface, and everything except the slope $\frac{dh}{ds} = \frac{i}{\sqrt{1-i^2}}$. For the line II

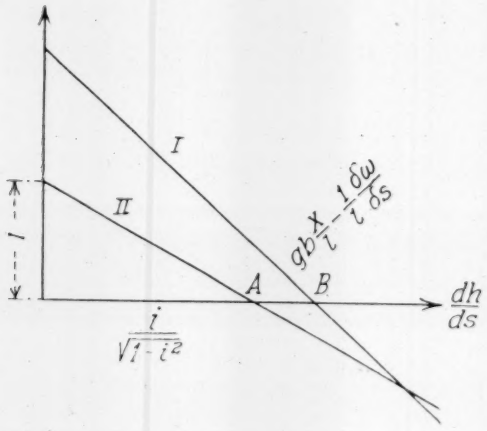

FIG. 2 VOLUME OF FLOW AND AXIS DIAGRAM

(second member) the shape of the section, elevation of water, volume of flow, and character of the river bed (roughness) are supposed to be given. The point of intersection at the axis of the abscissae is given by

$$bx - \frac{1}{g}\left(\frac{\delta\omega}{\delta s} + l\frac{dh}{ds}\right) = 0$$

or

$$\frac{dh}{ds} = gb\frac{x}{l} - \frac{1}{l}\frac{\delta\omega}{\delta s}$$

This point of intersection is independent of the volume of flow, but when the discharge increases, the ordinate of the origin increases also.

Use of the diagram. For a given volume of flow line II may be drawn; the intersection of I and II has for its abscissa the inclination of the axis for given elevation of the water surface and volume of flow. There is however such a value of the volume of flow q' for which the inclination of the axis is indefinite: this happens when I and II are parallel.

DISCUSSION OF THE DIAGRAM

First Case: $\frac{i}{\sqrt{1-i^2}} < gb\frac{x}{l} - \frac{1}{l}\cdot\frac{\delta\omega}{\delta s}$

When q increases, $\frac{dh}{ds}$ decreases, except in the case of the sudden jump from $-\infty$ to $+\infty$ for q_r.

Second Case: $\frac{i}{\sqrt{1-i^2}} = gb\frac{x}{l} - \frac{1}{l}\cdot\frac{\delta\omega}{\delta s}$

or $\frac{dh}{ds} = \frac{i}{\sqrt{1-i^2}}$: horizontality, independent of the volume of flow.

Third Case: $\frac{i}{\sqrt{1-i^2}} > gb\frac{x}{l} - \frac{1}{l}\cdot\frac{\delta\omega}{ds}$

When q increases, $\frac{dh}{ds}$ increases also, except in the case of a sudden jump from $+$ to $-\infty$ for q_r.

The regions of

$$\frac{i}{\sqrt{1-i^2}} > gb\frac{x}{l} - \frac{1}{l}\cdot\frac{\delta\omega}{ds} \quad \text{(positive region)}$$

and

$$\frac{i}{\sqrt{1-i^2}} < gb\frac{x}{l} - \frac{1}{l}\cdot\frac{\delta\omega}{\delta s} \quad \text{(negative region)}$$

may be considered to be separated by *equality curves*, and as a particular case only one region may exist, as is generally the case with prismatic channels.

The axes may for the purpose of this investigation be divided into continuous parts. Then (*a*) if an axis is in the positive region, $\frac{dh}{ds}$ increases when q increases, except in case of a passage from $+\infty$ to $-\infty$; (*b*) if an axis is in the negative region, the opposite happens; (*c*) when an axis intersects an equality curve, it is horizontal.

Corollaries: if two axes are in the same region: (*a*) they can intersect only in one point; (*b*) the axis of the larger volume of flow is above the other downstream from the point of intersection, and below it upstream, if the region is positive, and vice versa. If two axes are located in two consecutive regions, they can have not more than two points of intersection; if they have a common point on the equality curve, they are tangent to it and can have no other common point.

Internal Combustion Engines

UTILIZATION OF EXHAUST HEAT FROM INTERNAL-COMBUSTION ENGINES (*Abwärmeverwertung bei Verbrennungskraftmaschinen*, K. Kutzbach, *Stahl und Eisen*, vol. 33, no. 28, p. 1133, July 11, 1912. 2½ pp., 6 figs. *d*). Discussion of the problem of exhaust heat utilization in connection with internal-combustion engines. This problem is more complicated and less

attractive than the similar problem with steam engines. Comparatively a much smaller proportion of heat can be saved in the first instance (Fig. 3). The heat carried off by cooling water can be saved only partly in extremely rare cases, especially in large gas engines, because if the temperature of the water exceeds 40 deg. cent. (104 deg. fahr.) there is danger of pre-ignition. The exhaust gases cannot be used directly, like steam, because with their large volume they have a comparativly low heat content and give up their heat about 200 times slower than steam, and they contain as a rule various impurities, such as oil, sulphuric acid, and soot. Their only uses therefore appear to be for the production of steam for heating purposes, power production, and for gas producers. The author points out that where gas producers use fuels free of tar the waste heat may be utilized for making all the steam which they require, but when the fuel contains tar this cannot be done because tar rapidly destroys

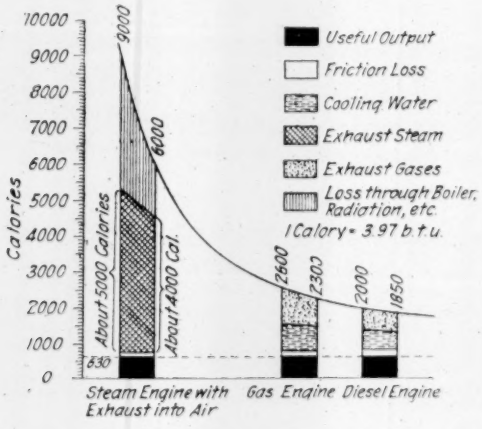

Fig. 3 Heat Balances of Steam, Gas and Diesel Engines

the boilers. The article contains a description of the horizontal and vertical types of boilers for utilizing the exhaust heat of internal-combustion engines constructed by the Maschinenfabrik Augsburg-Nuremberg. Vertical boilers are generally used for the production of hot water, horizontal for steam making.

Efficiency of the Explosion Gas Turbine (*Der Wirkungsgrad der Explosions-Gasturbine*, H. Holzwarth, *Zeits. des Vereines deutscher Ingenieure*, vol. 56, nos. 24 and 25, June 15 and 22, 1912. 8 pp., 5 figs. cc). An article of controversial nature in which the author attempts to prove that his formulae and data are on the whole preferable to those of Professor Stodola as far as the calculation of the efficiency of the explosion gas turbine is concerned. He shows further that the heat processes in the gas turbine can be investigated only by graphical methods if heat losses and variations in the specific heats are to be taken into consideration: analytical methods are said to be entirely inapplicable for this pur-

pose. The article also contains valuable data of tests on gas turbines made by the author since the publication of his book. The same journal, No. 25, June 22, contains an answer by Professor Stodola in which he defends his views and attempts to show that the gas engine cannot absolutely have as high an efficiency (30 per cent) as Mr. Holzwarth claimed for it. For the gas entropy table discussed in an appendix to Professor Stodola's article see p. 1397.

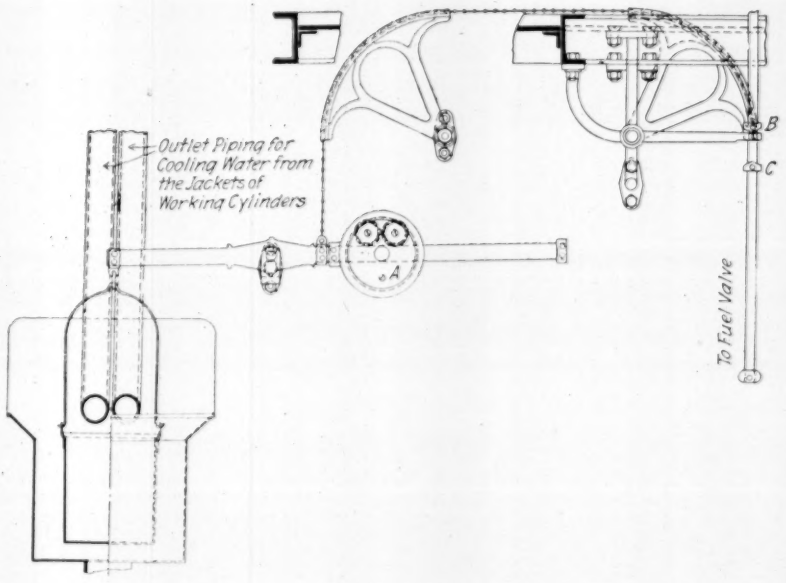

FIG. 4. J. A. STROGANOFF DEVICE FOR STOPPING AN INTERNAL-COMBUSTION ENGINE WHEN COOLING WATER CEASES TO CIRCULATE

AUTOMATIC DEVICE FOR STOPPING INTERNAL COMBUSTION ENGINES IN CASE OF LACK OF COOLING WATER. Designed by J. A. Stroganoff of Twer, Russia (*Selbsttätige Anhaltsvorrichtung für Verbrennungsmaschinen bei Kühlwassermangel, Bauart J. A. Stroganoff* (Twer), N. Bikoff and G. v. Doepp. *Die Gasmotorentechnik*, vol. 12, no. 4, p. 62, July 1912. *d*). Description of a safety device intended to prevent overheating of an engine in case the supply of cooling water is cut off without being noticed by the engineer. A valve in the fuel piping of the motor is connected with one arm of a lever having on the other arm a vessel into which flows water from the cylinder jackets. When everything is in order, part of the water flows over the edge of the vessel while the rest escapes through little openings in the bottom of the vessel which, being full, occupies the lowest position possible with the prolongation *B* in Fig. 4, but when the flow of water ceases, the vessel empties, and the counterweight *A* moves up as far as *C* permits it, causing the valve in the fuel piping to be closed and the admission of fuel to the motor to be shut off.

ALBARET GASOLENE MOTOR FOR FARM WORK (*Moteur à essence Albaret pour l'agriculture. Le Génie Civil*, vol. 61, no. 11, p. 226, July 13, 1912. 1 p., 3 figs. *d*). Description of the Albaret farm gasolene motor built by the Société des Anciens Établissements Albaret, of Rantigny, Oise, France. As shown in Fig. 5, the cylinder *C* and its water jacket are cast in one piece with the crank case *C'* provided with the opening *P'* for the inspection of the inside. By means of a connecting rod *b*, the piston *P* operates a handle *m* located at one of the extremities of the shaft *a* carrying at its other end the pulley-flywheel *V*; this shaft rotates inside a long and strong bearing supported by the crank case *C'*, the flywheel *V* being given a curved shape in order to bring the plane of the center of gravity inside the bearing. Inside the crank case *C'* is placed the fuel tank *R*, the

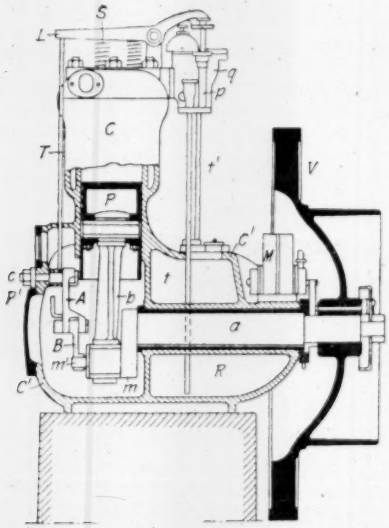

FIG. 5 ALBARET GASOLENE FARM MOTOR

gasolene passing through the pipe *t* and pump *p* to the carburetor *q*, any excess of the fuel returning to the tank by the overflow pipe *t'*. There is no special lubricating apparatus, a suitable quantity of oil being simply poured into the crank case where the handle *m* is placed, with the level indicated by a tube outside. A high tension magneto takes care of the ignition, the timing being automatically regulated.

The article describes also the system of governing which is of the hit-and-miss type, but rather complicated.

GERMAN AUTOMOBILE DESIGN (*Deutsche Automobilkonstruktionen, Auto-Technik*, vol. 1, no. 13, appended to the Allgemeine Automobil-Zeitung, vol. 3, no. 29, p. 33, July 19, 1912. 6 pp., 9 figs. *d*). Description of the Sperber 4-cylinder 18-h.p. automobile. The Sperber carburetor is shown

in Fig. 6. The gasolene enters through 1, and is regulated by a float spindle in the usual manner. From the float chamber it passes through duct 2 into a ring-shaped groove in the female cone of clutch 3 provided along its inner circumference with fine nozzle-shaped openings; the gasolene squirts through these openings and forms an explosive mixture with the air coming from below. The mixture is regulated by the vertical displacement of the cone 4 which is free to move on the spindle of the throttle valve 5,

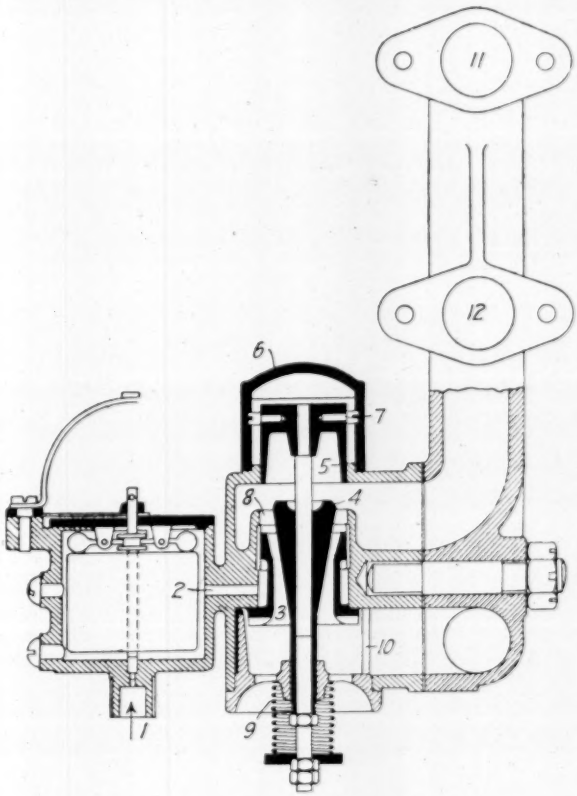

Fig. 6 Sperber Carburetor

and can be raised both by hand and by suction. The throttle valve may be regulated either from the accelerator pedal by a regulating rod, or by turning the bell-shaped case 6 from a lever on the steering wheel. The casing 6 is provided with spiral grooves in which run little screws 7 set in the throttle valve, and when the casing turns, the screws force the valve to move up and down; when the opening is throttled entirely, the valve goes down into the cylindrical guide 8. The air ratio of the mixture is regulated by the rotary slide-valve 9 varying the opening 10

for the admission of air (preheated). Owing to automatic air regulation, the carburetor, with the throttle valve open adapts itself to the speed of the motor, and delivers a correct mixture as well at high speed as at low. The bend connecting the pipe to the right of the carburetor contains two pipes leading in opposite directions: the first for warm air goes from 11 at the top down to the lower part of the carburetor, and passes on the way through some of the hot water around the cylinder casting, so as to preheat the air. The other flange 12 is screwed on to the suction piping. The whole suction piping is cast in one piece with the cylinder casting.

INVESTIGATION OF THE WORKING PROCESS OF A TWO-CYCLE MOTOR (*Untersuchung des Arbeitsprozesses eines Zweitaktmotors*. H. Scheit and

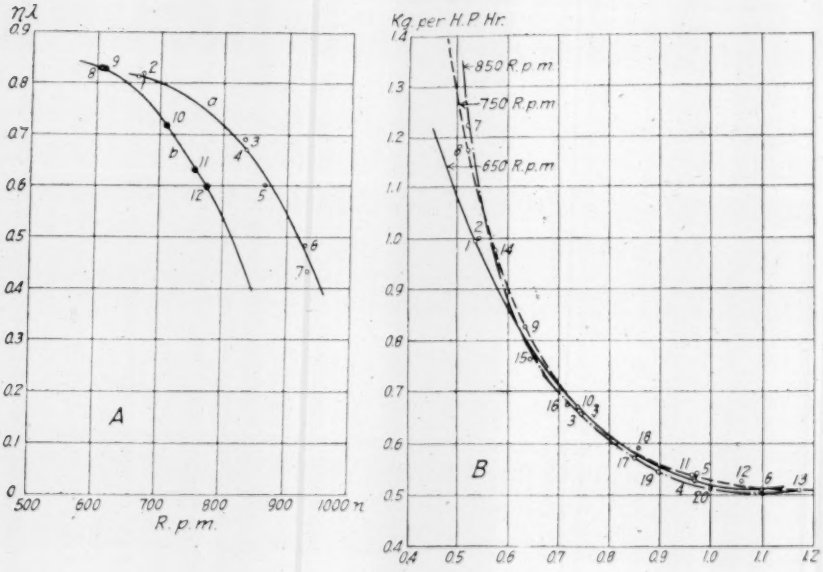

FIG. 7A EFFICIENCY OF PUMP AND SPEED OF TWO-CYCLE MOTOR
FIG. 7B RATIO OF MIXTURE AND FUEL CONSUMPTION OF TWO-CYCLE MOTOR

Assistant Bobeth. *Zeits. des Vereines deutscher Ingenieure*, vol. 56, no. 22, p. 862, June 1, 1912. 8 pp., 18 figs. c.A). Account of tests of a two-cycle motor made at the Royal Saxonian Mechanical and Technical Laboratory in Dresden, with a view of determining the working process of two-cycle motors. The motor tested was a single-cylinder motor-boat engine of the Ferro Machine and Foundry Co., of Cleveland, O., rated at 4 h.p., 800 r.p.m., 3¾ in. bore, 3½ in. stroke, jump spark ignition. Gasolene was used as fuel with an average upper heating value of 11054 WE (19,945 b.t.u. per lb.), and lower heating value 10334 WE, or 18,660 b.t.u. per lb. From the chemical analysis of the fuel it was found that 3.483 kg of

oxygen were required for the combustion of 1 kg of fuel, this amount corresponding to 14, 91 kg of air, or at one atmosphere pressure and 15 deg. cent. (59 deg. fahr.) temperature 12.55 cbm (200 cu. ft. of air per lb. of combustible).

The first series of tests had the purpose of determining the relation between the efficiency of the fuel pump and speed of rotation of the motor. To reduce the resistance in the suction piping the throttling valve at the carburetor was thrown wide open, and the most favorable opening of the fuel nozzle and position of the spark determined for each speed by previous tests. The tests as shown in curve *a* (Fig. 7 A) prove that the efficiency of the pump at slow speeds is pretty high, but rapidly falls with the increase in the number of revolutions; this may be due either to the increase of the resistance in the suction piping as the velocity of air increases, or to oscillations in the sucked-in column of air. The fact that the inlet opening is uncovered only at the end of the stroke and that the length of time while it is uncovered decreases as the speed of the motor increases may be of importance. A motor with the inlet opening governed not by the piston, but by a valve in such a manner as to be open during the whole suction stroke would therefore be more efficient, provided it were simple in construction.

Curve *b* shows the efficiency of the pump as a function of the speed of the motor when the gasolene was shut off, and the motor driven by an electric motor worked only as a pump. The purpose of this test was to eliminate the influence of the resistance *s* in the exhaust and at the entrance of the charge into the working cylinder. The comparison of the two curves shows that in the second case with a constant back pressure equal to atmospheric, the efficiency decreases more rapidly with the increase in speed. It may be that during regular working of the motor the pressure in the cylinder at the end of expansion falls below atmospheric and the flow of exhaust gases produces a supplementary suction: this could not be definitely established by means of an indicator.

Influence of Ratio of Mixture on Gasolene Consumption. Three series of tests were made, with the motor working at about ¾ of its maximum output, and at 650, 750 and 850 r.p.m., with the indicated ratios of mixture, the best position of the spark for each ratio of mixture having been found by previous tests. The three lines (Fig. 7 B) show how substantially the fuel consumption rises with an insufficient supply of air. The lines for the three speeds of rotation nearly coincide, showing thereby that the speed of rotation has practically no influence on fuel consumption. A curve drawn by combining these three curves has shown that the most economic way of running the motor is with a 10 per cent excess of air.

Experiments were also made on the influence of air preheating, spark displacement, and cooling of exhaust gases. In general the tests have shown that the minimum fuel consumption of the Ferro two-cycle motor is 500 g (1.1 lb.) per h.p.-hour. This is more than the fuel consumption of a four-cycle motor of equal power which appears to be due partly to some fresh fuel being carried away by the exhaust gases and partly to a poorer ratio of mixture. The average effective piston pressure is also equal to

only about one-half that in the four-cycle motor. The charging pump, though found to be very efficient absolutely, appears to be not strong enough in the case of a two-cycle motor to produce an effective scavenging of the cylinder, this task being made particularly hard by the fact that the time for it, which is only a fraction of the piston stroke, is very short.

The efforts of designers of two-cycle motors must therefore be directed towards improvement of the charging process. The most promising way appears to be the separate introduction of air and fuel, and the use of excess of air for scavenging.

Machine Shop

DOUBLE MILLING MACHINE FOR TORPEDO SCREW BLADES OF J. E. REINICKER IN CHEMNITZ-GABLENZ (*Doppelte Fräsmaschine für Torpedo Schraubenflugel von J. E. Reinicker in Chemnitz-Gablenz*, F. Nickel, *Zeits. des Vereines deutscher Ingenieure*, vol. 56, no. 23, p. 929, June 8, 1912. 3 pp., 6 figs. *d*). The machine mills the blade from a template, cutting in strips. The table carrying the screw has an automatic quick return and a stop motion which is thrown into gear upon completion of the work.

BROAD OR NARROW WELDED SEAMS (*Breite oder schmale Schweissnahte. Autogene Metallbearbeitung in Acetylen*, vol. 15, no. 12, 1 p. *c*). The Technical Bureau of the Association for Autogenous Welding in reply to inquiries as to whether or not there is any technical advantage in making broad welded seams, meaning by this term melting the metal at a considerable distance on both sides of the welded joint, stated that the advantage was doubtful in most cases. Melting considerable amounts of metal in the immediate proximity of the seam and thoroughly joining it with the welded-in material helps to prevent oxidized particles from getting incrusted in the joint, but on the other hand subjection of the metal at the seam to prolonged action of the flame produces changes in its structure, and, owing to the high heat conductivity of metals, leads to considerable heat losses. A better way is to heat by a large burner the metal in the neighborhood of and parallel to the seam: this permits the heat to concentrate at the seam when a comparatively small burner is sufficient to make a good joint even in the case of thick plates, there being less possibility of overheating the metal than with the broad seam. In welding thick plates when the material along the seam is heated by a large burner to a dark cherry red color, the welding proper proceeds very rapidly without any need for melting large amounts of material around the seam.

Mechanics

FRICTION GEAR WITH VARIABLE RATIO OF SPEED (*Transmission par friction à rapport de vitesses variable*, P. Raes. *Annales de l'association des ingénieurs sortis des écoles spéciales de Gand*. Ser. V, vol. 5, no. 2, 1912. p. 188, 18 pp., 9 figs. *td.1*). Theoretical discussion of the principles of transmission by friction, particularly the problem of finding by calculus rolling surfaces of such a profile that by sliding along perpendicular axes they should produce simple rolling. The author shows that theoretically this condition is satisfied if one of the solids is generated by a parabola of the second degree rotating about its axis of symmetry, while the other

solid is generated by a semi-parabola of a double parameter rotating about a tangent at the vertex. There are however considerable difficulties in embodying this principle in a variable-speed transmission in a small automobile, and further the author shows that the condition of simple rolling is not of sufficient importance to justify a very complicated system of transmission. The author further analyzes the action of a disk friction gear, such as is used in some actual automobile construction.

Materials of Construction

TESTS OF BOILER PLATES WITH LAP WELDED JOINTS (*Versuche mit überlappt geschweissten Kesselblechen*, B. Zwiauer, *Zeits. des Vereines deutscher Ingenieure*, vol. 56, no. 22, p. 877, June 1, 1912. 4 pp., of which two are tables, 5 figs. *e*). Tests of lap welded joints, mainly with a view to the existing German regulation for the inspection of boilers.

Steam Engineering

ELECTROLYTIC OIL SEPARATION (*Elektrolytische Oelabscheidung*, W. Heym. *Elektrochemische Zeits.*, vol. 19, no. 4, p. 21, July 1912. 3 pp. *td*). Oil is carried away by the exhaust steam partly in the form of a finely divided liquid, partly in that of vapor, a considerable amount of oil or some of its constituents being vaporized no matter what is its boiling point. However well oil separators working on the mechanical principle be arranged, they can catch practically only the oil present in the form of a spray, about 75 per cent of the total, while the rest escapes and reaches the condenser. It is there transformed by the action of the water into minute spheroidal corpuscles, and is partly carried off with the feed-water, partly settles on the condenser walls, and helps to form a very unsoluble crust. The properties of this crust are determined by the fact that these oil corpuscles are charged with electricity. The crust can therefore be treated just like every other colloidal solution, and the oil particles may be settled down by being acted upon by a colloidal solution carrying an opposite charge, e. g., by colloidal iron hydroxide. On this property is based the *oil separating process* of Davis-Perret. It consists mainly in electrolysing the water of condensation by means of iron electrodes, there being added to the water some electrolyte, usually sodium carbonate, to make it more conductive. A basic salt is formed, and the alkalies settle at the anode, the oil particles being caught in this precipitate, and practically entirely separated out by subsequent filtration. The efficiency of the Davis-Perret oil separating process has been found to be as high as 98.5 per cent. It is further stated that since by this process a precipitate is formed which is not a pure hydroxide, but a basic salt, a certain excess of it reaches the boiler and makes the water alkaline; this in its turn makes the ability of the water to decompose iron lower than with pure water, owing to the fact that the ability of water to affect iron is proportional to the hydrogen content of the water. The article also describes how a simplified type of Richard's nepthelometer may be used for the determination of the contents of oil in water.

NEW PATENTS IN THE DOMAIN OF STEAM BOILER FIRING (*Neue Patente auf dem Gebiete der Dampfkesselfeuerung*, Pradel. *Zeits. für Dampfkessel und Maschinenbetrieb*, vol. 35, no. 30, p. 315. 2 pp., 10 figs. *d*). Brief accounts of new German patents granted during the first half of 1912 for appliances in steam boiler firing, mainly grates.

ON THE EXPANSION OF SATURATED STEAM (*Sur la détente de la vapeur d'eau saturante*, A. Leduc. *Comptes rendus de l'Académie des Sciences*, vol. 155, no. 1, p. 33, July 1, 1912. 2 pp. *tA*). The object of the author's investigation was to find the precise value of n in Rankine's formula: $pv^n = \text{const.}$ for the expansion of dry saturated steam on the basis of numerical data obtained by Holborn and Henning and Dieterici, and the author's own formula of state for real gases

$$Mpv = RT\phi$$

where M is the molecular mass of a gas, R^* an absolute constant common to all gases; ϕ is called by the author the molecular volume of a gas, and is the ratio between the relative (say with respect to air) molecular weight of a gas to its relative density.

Let a unit of mass of the fluid pass through Rankine's cycle between T_0 and T_1, and let x be the quality of the steam at the end of adiabatic expansion. If k be the specific heat of water at T deg. under the pressure of saturated steam, and L its latent heat of evaporation, then

$$\int_{T_0}^{T_1} k\frac{dT}{T} + \frac{L_1}{T_1} = \frac{L_0}{T_0}x$$

A unit of mass of steam occupies a volume v_1 at T_1 under the maximum pressure p_1, while at T_0 it is only the mass x that occupies the volume v_0 under the maximum pressure p_0 (the volume $1 - x$ of the condensed water being negligible). Hence

$$M\ p_1\ v_1 = RT_1\phi_1$$

and

$$M\ p_0\ v_0 = xRT_0\phi_0$$

from which

$$\frac{x\ p_1 v_1}{T_1\ \phi_1} = \frac{p_0 v_0}{T_0\phi_0}$$

which combined with

$$p_1 v_1^n = p_0 v_0^n$$

gives

$$\frac{n-1}{n} = \frac{\log T_1 - \log T_0 + \log \phi_1 - \log \phi_0 - \log x}{\log p_1 - \log p_0}$$

Table 1 has been calculated for intervals of 10 deg.

* In a previous investigation (Annales de Chimie et de Physique, septième série, tome 15, p. 92) the author found that R in C.G.S. units, with the molecular mass of oxygen taken as 32, is equal to 8319×10^4, or in centimeters of mercury $R' = 6237.2$. He refers in a footnote for the notation in the present article to his former article in the *Comptes rendus*, vol. 153, p. 51, which does not contain it. The reader is referred to the above quoted article in the *Annales*.—EDITOR.

TABLE 1 PROPERTIES OF SATURATED STEAM FOR TEMPERATURES BETWEEN 160 AND 80 DEG. CENT. (320 TO 176 DEG. FAHR.)

Deg.	x	n	Deg.	x	n
from 160 to 150	0.9829	1.142	from 120 to 110	0.9810	1.142
from 150 to 140	0.9824	1.144	from 110 to 100	0.9805	1.140
from 140 to 130	0.9820	1.144	from 100 to 90	0.9800	1.137
from 130 to 120	0.9815	1.144	from 90 to 80	0.9794	1.134

The article contains also an additional table showing that slightly different results are obtained with the Regnault data.

The value of n passes through a maximum at about 135 deg. Nevertheless Rankine's formula may be used with n constant, provided its value be taken as 1.14 when the temperature of steam varies between 160 and 60 or even 40 deg. cent. (320 to 140 to 104 deg. fahr.), and as 1.135 when the engine works with a low pressure and has a very well cooled condenser. With the fall of initial temperature x decreases. From Table 1 it may be seen that x, after an expansion of θ deg. from the temperature of 160 deg. cent. may be exactly represented by

$$x = (0.9983)^{\theta}\left[1 - \frac{\theta(\theta-1)}{2} \times 5 \times 10^{-6}\right]$$

but in the case of steam engines practically for all initial temperatures the following simpler formula may be used:

$$x^1 = (0.9981)^{\theta}$$

IMPROVEMENT OF STEAM TURBINE EFFICIENCY BY THE USE OF "EXHAUST DISKS" (*L'amélioration du rendement des turbines à vapeur par l'emploi des "disques d'échappement,"* Delaporte, *Revue de mécanique*, vol. 30, no. 6, p. 517, June 30, 1912. 7 pp., 7 figs. *ed*). The last row of blades has always long blades as compared with the thickness of the distributed steam films, and the efflux comes out in an indefinite film, that is without the separate stream lines being given an opportunity to expand in the direction of the axis of a blade. Owing to the immediate proximity of other vapor films, the films are not allowed to expand in the perpendicular direction *ab* (Fig. 8, A) either. The guiding *ac* prevents entirely any change of the direction of the films in the triangular space *abc*: there can be therefore no expansion, and the pressure P at *ab* is constant for the entire space, and particularly for the section below disk *bc*. As a result, the difference between P and the pressure p at the plane of outlet of the disk and at the exhaust gives only to the steam an acceleration A parallel to the axis of the disk; the change in direction produced affords the films the possibility to thicken in accordance with the increase of the relative velocity of the steam from V to V', but as far as useful work is concerned this acceleration is entirely lost, because it is not accompanied by a change in the moment of the quantity of motion with respect to the

axis. *There is therefore a certain limited pressure beyond which the turbine ceases to benefit by any further increase of expansion.* This limiting pressure P' is characterized by the relative velocity at the outlet of the disk being 360 meters (1181 ft.) per second, the output being

$$\triangle = 0.97\, Ps$$

where $\triangle$ is in kg per hour, and s the total section open to the passage of the steam in sq. mm. (1 sq. mm. = 0.00155 sq. in.), Hence

$$P = 1.03\, \frac{\triangle}{s}$$

The fact that there are many correct graphical proofs of the important advantages derived by turbines from reducing the aval pressure does not

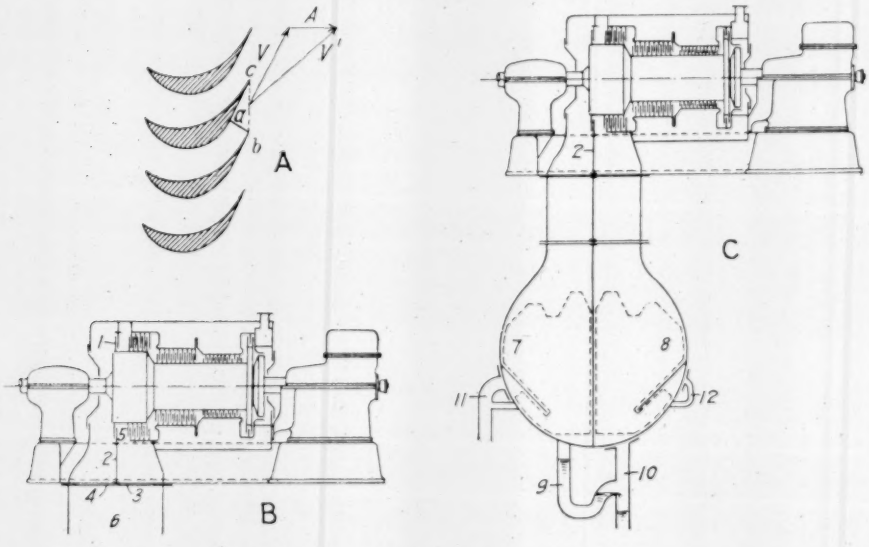

Fig. 8 Delaporte Exhaust Disks and Divided Condenser

conflict with the above statement: all such proofs hold only for cases where the ratio $\frac{\triangle}{s}$ is small and the pressure P' is below the pressure p of the condenser. But with the increase in the size of turbines which for economic reasons has not been followed by a reduction of their speed, the output $\triangle$ has increased, while the section s could not be increased at the same rate owing to its being limited by the strength of the materials used. A disk of 750 mm (say 30 in.) in diameter, with blades 100 mm (3.9 in.) long, and a coefficient of distribution λ' from 0.6 to 0.65 ($\lambda' = \frac{ab}{bc'}$, Fig. 8A) represents about the limit of what can be done at 3000 r.p.m.

without sacrificing the safety of operation, and that gives to s a maximum value of about 150,000. The ratio $\frac{\triangle}{s}$ thus becomes an element which can no longer be neglected.

As an example the author considers a turbine driving a 1500-kw. alternator at 3000 r.p.m. With average conditions of pressure, superheat and vacuum the steam consumption is 6.9 kg (15.18 lb.) per kw. per hour; hence $\triangle = 6.9 \times 1500 = 10{,}350$ kg, and

$$P = 1.03 \frac{\triangle}{s} = 1.03 \times \frac{10350}{150000} = 0.71 \text{ kg} = 1.009 \text{ lb.}$$

If the cooling condenser water be taken at 16 deg. cent. (60.8 deg. fahr.), and comes out of the condenser at 28 deg. cent. (82.4 deg. fahr.), the pressure p at the entrance to the condenser will be approximately that corresponding, according to steam table, to a temperature of about 5 deg. above the latter temperature, or $p = 0.051$ kg $= 0.729$ lb. The expansion from 0.071 to 0.051 kg is lost. But if a disk with $s = 150{,}000$ cannot take care of the entire steam at a pressure below 0.071 kg, another disk with equal section could drive off the steam at the pressure of 0.051 kg, if it had only to take care of a reduced amount of steam: this is the fundamental idea of the "exhaust disk." It is provided, like others, with a distributor, but this is placed between the two exhausts, and is traversed only by a part of the total flow of steam, the inlet connecting branch being contracted so as to limit the pressure in the main turbine to a value beyond which it ceases to be of benefit. Fig. 8 B gives an example of this arrangement: 1 is the exhaust disk of which the distributor seats on the cast-iron plate 2 between the branches 3 and 4, the first of which is contracted by a diaphragm (it is often more convenient to create a loss of pressure at 5). The two streams of steam reunite in 6 in the exhaust pipe. If five-eighths of the flow of steam be sent through the exhaust disk, 88 h.p., or about 3.9 per cent of the normal power, is recovered, which is significant considering the simplicity of the device. This gain can be still further improved by modifying the condenser arrangement somewhat. As shown in Fig. 8 C, the partition 2 is extended through the exhaust pipe right into the condenser which it divides into two parts. With regard to the circulation of water, the nests of tubes 7 and 8 are disposed in series with 7 receiving the fresh water. As a matter of fact this arrangement results in there being practically two separate condensers of which one, viz. that receiving the steam from the exhaust disk (the author calls it the cold condenser) permits of obtaining a final pressure p' considerably below the pressure p of the other condenser, or the one that would prevail throughout the entire condenser if it were not divided. Notwithstanding the difference of internal pressures, both condensers may be supplied with water by the same pump. With the modified condenser an increase in efficiency up to 6.8 per cent has been obtained in actual tests. The article contains full data of five tests.

Boiler Feed Regulator, Degrémont system (*Régulateur d'alimentation, système Degrémont*, F. D. *La Technique moderne*, June 15, 1912. ½ p., 1 fig. *d*). The Degrémont boiler feed regulator (Fig. 9), consists of a body A provided with a membrane *m*, and connected with the boiler by the pipes, *e*, *f*, *g*, and *h*, in such a way that the pipe *h* reaches the average level of water in the boiler. As long as *h* is in the water, the entire system of piping is filled by water, and equal pressures act on both sides of the membrane. But when the water in the boiler goes below the end of the pipe *h*, the water in this pipe runs out and the pipe is filled by steam. As a result, the pressure on the upper side of the membrane becomes predominant, owing to the additional pressure of the column of water in the pipe *g*, and the membrane is deflected downwards, and in doing so displaces by means of a suitably arranged lever the counterweight *b* acting on a flap valve *c* admitting feed water to the boiler. The admission of water continues until the water level rises above the lower end of the pipe *h*. This apparatus was shown at the Roubaix Exhibition of 1911.

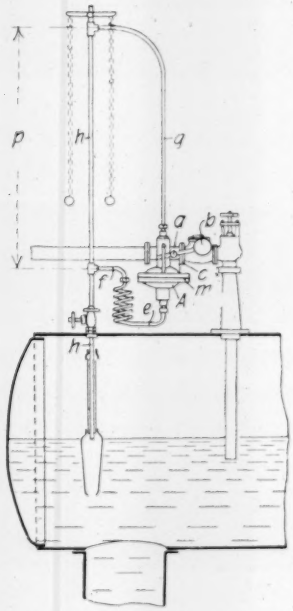

Fig. 9 Degremont Feedwater Regulator

Utilization of High Vacua in Turbines with Small Cross-sections of Outlet (*Die Ausnutzung hoher Luftleere in Dampfturbinen bei kleinen Austrittquerschnitten*, F. Lösel. *Zeits. des Vereines deutscher Ingenieure*, vol. 56, no. 25, June 22, 1912. 8 pp., 15 figs. *cA*). Description and data of an investigation undertaken to determine whether in turbines

having cross-sections of outlet dimensioned for vacuum of 90 per cent higher vacua from 95 to 96 per cent may be utilized, and how the data obtained may be reduced to the basis of a vacuum obtained with a given temperature of cooling water of 25 deg. cent. (77 deg. fahr.: this was the temperature at which the nominal efficiency of the turbine tested was guaranteed by the manufacturers). The turbines were designed on the assumption that 90 per cent vacuum was reached without fail. The tests have shown: (*a*) that high vacua from 95 to 96 per cent may be fully utilized, and the output of the turbines grows with the vacuum; (*b*) that the thermodynamic efficiency referred to the state in front of the first diaphragm is the same for all vacua from 90 to 96 per cent, which shows further that the heat is well utilized, notwithstanding the small cross-section of outlet, even with the highest vacua. It is therefore quite permissible to accept the same efficiency in reducing the steam consumption to better vacua (the author states that this was denied by previous investigators); (*c*) the fact that the output of a turbine grows with the increase of the vacuum while the thermodynamic efficiency does not change may be explained only by the assumption that the velocity of the steam increases beyond the critical velocity, and that this high velocity is utilized. Since the pressure in the stage next to the last is the same for all vacua, it is only the last two stages that take part in the utilization of the high vacua. With a vacuum of 90 per cent for which the turbine was designed, already in the stage next to the last the pressure falls to that of the condenser, and the last stage not only produces no work, but consumes some through blade friction. The stage next to the last therefore has the best thermodynamic efficiency. The article contains complete data of these tests.

STEAM MACHINERY AND FIRE ENGINES OF LEONARD DA VINCI (*Dampfapparate und Feuermaschinen bei Leonardo da Vinci. Zeits. des Oesterr. Ingenieur und Architekten Vereines*, vol. 64, no. 28, p. 440, July 12, 1912. 2 pp. 7 figs. *h*). Description of a steam-gun and other steam machinery invented or described by Leonardo da Vinci, and particularly some experiments made by him to determine the volume and pressure of steam; also description of fire engines invented by him.

Thermodynamics

GAS ENTROPY TABLE FOR THE LATEST VALUES OF SPECIFIC HEATS (*Gasentropietafel für die neusten Werte der spezifischen Wärmen*, A. Stodola. *Zeits. des Vereines deutscher Ingenieure*, vol. 56, no. 25, p. 1008. 2 pp., 3 figs. *tA*.). The author bases his new table on the values of Pier as given in the German engineering pocket-book "Hütte" (in kilogram-molecules per deg. cent.):

Steam $C_v = 6.065 + 1.0 \times 10^{-3} t + 0.2 \times 10^{-9} t^3$

Carbon dioxide $C_v = 6.80 + 3.3 \times 10^{-3} t - 0.95 \times 10^{-6} t^2 + 0.1 \times 10^{-9} t^3$

which he replaces, for the purposes of the entropy table, by equations of the type (in kg.-molecules per deg. absolute):

$$C_v = a_v + bT + cT^2 \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots [1]$$

where $a_v = 4.67$, and b and c are so selected that the variations within the technically important range of temperatures, from 500 to 2000 deg. cent. absolute, be as small as possible. The above formulae become then

For steam $C_v = 4.67 + 1.263 \times 10^{-3} T + 1.18 \times 10^{-6} T^2$

For carbon dioxide $C_v = 4.67 + 8.004 \times 10^{-3} T - 2.337 \times 10^{-6} T^2$

which differ but very little from the values derived from the Pier equations, as shown by Fig. 10A. From C_v the specific heat for constant pressure may be obtained by the usual formula

$$C_p = AR + C_v \text{ or about } 2 + C_v \quad \ldots\ldots\ldots\ldots\ldots\ldots [2]$$

The introduction of [1] and [2] into the formula of entropy in the case of gases gives

$$S = a_v \lg n \left(\frac{T}{T_o}\right) + b(T - T_o) + \frac{c}{2}(T^2 - T_o^2) + AR \lg n \left(\frac{V}{V_o}\right) =$$

$$a_p \lg n \left(\frac{T}{T_o}\right) + b(T - T_o) + \frac{c}{2}(T^2 - T_o^2) + AR \lg n \left(\frac{p}{p_o}\right)$$

which is the old Stodola formula of entropy for gases with the addition of the

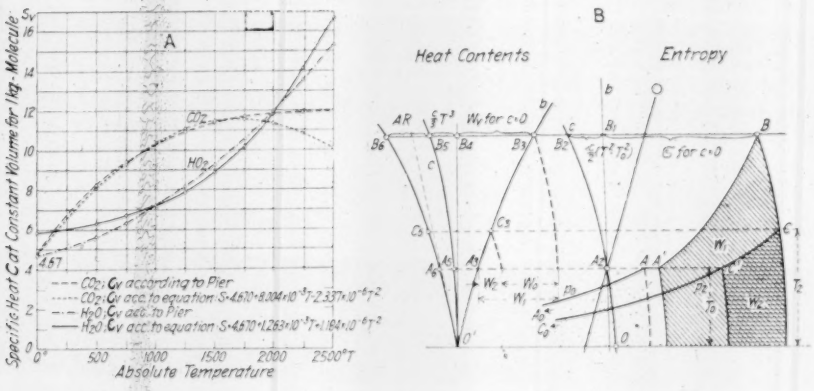

Fig. 10A Specific Heats of Carbon Dioxide and Steam

Fig. 10B Entropy Diagram for Specific Heats Varying according to Parabolic Law

term containing c, and it is sufficient to plot on the line b in the entropy table the parabola $\frac{c}{2}(T^2 - T_o^2)$ from Fig. 10B, to obtain, within the range BB_2 the new entropy values. It is to be measured from the curvilinear axis c and gives as an adiabatic curve, i. e. with $S = \text{const.}$, the c parabola itself only displaced about BB_2, as indicated by the expansion line BC. This adiabatic curve preserves the property according to which the change of the heat content during a reversible change of state is equal to the area between the curve of state and the adiabatic curves down to the zero line.

The internal heat energy is expressed by

$$W_v = \int_o^T C_v d\,T = a_v T + \frac{b}{2} T^2 + \frac{c}{3} T^3$$

In the entropy diagram, B_3B_4 is for the heat energy W_v for $c = 0$; if the cubic parabola $\frac{c}{3} T^2$ be constructed from $0'$, B_3B_5 will represent the total heat energy.

For the heat contents the author gives the expression

$$W_r = \int C_p dT = \int (C_v + AR) dT = W_v + ART$$

As an illustration the author constructs an entropy diagram for the Holzwarth gas turbine.

Miscellanea

MOTOR PLOWS TESTING (*Die Prüfung von Motorpflügen*, Dr. Martiny. *Der Motorwagen*, vol. 15, no. 20, July 20, 1912. *t*). The object of testing is the determination of the agricultural value of a motor plow. The agricultural value of plows with stationary cutters, such as the Stock and I.H.C. types, depends on the total cost of plowing a unit of ground (as compared with plowing by horses, steam, or electricity), cost of additional operations, and quality of work; that of plows with rotating cutters, such as the Köszegi type, depends also, perhaps primarily, on the influence of the peculiar handling of the earth, by this kind of plow: finely cutting and thoroughly mixing it. The total cost of plowing a unit of ground depends on the cost of upkeep of the machine (depending in its turn on the first cost, durability and strength of the plow), on the cost of fuel and attendance, and the amount of ground plowed per unit time to a given depth, apart from interruptions independent of the action of machinery. The output in units of plowed area cannot be determined just by measuring it, because it depends on the properties of the ground as well. (The article explains this in detail.) The efficiency of a motor plow depends further on the strain to which the plow is subjected at the tests. This may be done in two ways: either, while plowing, the gasolene consumption per hour, or number of revolutions per minute, and motor governing may be determined, or the turning moment of the engine may be measured by a torsion dynamometer; in laboratory tests the motor may be placed on the brake stand, and the hourly consumption of fuel, speed of rotation and governing may be determined.

In the case of motors with rotating cutters the most important are comparative tests of production. On the whole, tests of plows are not pure engine tests, but pass into the domain of plant life, in determining the agricultural result of the work delivered by the engine. This makes this kind of testing very difficult, complicated, and subject to errors which have not yet been quite elucidated: a statistical investigation parallel to the experimental appears therefore advisable.

LUBRICANTS (*Analysen von Motor-Schmiermitteln*, N. Bikoff and G. v. Doepp. *Die Gasmotorentechnik*, vol. 12, no. 4, p. 63. *e*). Several tables containing data of tests of various lubricants for internal-combustion engines made at the International Exhibition of Internal Combustion Engines in St. Petersburg. Most of the oils tested were of Russian manufacture.

PATENT LAWS OF HOLLAND (*Das holländische Patentrecht*, R. Bauch. *Zeits. fur Elektrotechnik und Maschinenbau*, vol. 15, no. 29, July 17, 1912. 2 pp. *g*). The first patent law went into force in Holland on June 1,

1912, and requires the patent application to be presented in five copies. The patent office considers only whether the description and drawings are such as to make the invention clear to men familiar with the respective art. The question of novelty is not considered, but due notice of the application is given, and the application itself may be seen in Holland and its three great colonies by anybody interested who may subsequently make objections to the grant of the patent. A term of six months is set for objections, and a small fee must be paid, the purpose of the fee being to prevent people from making groundless objections. The application must be made by the inventor himself or his attorneys. If the invention is not worked during five years the patent may be cancelled by the patent office unless there is a good excuse for the default. Three years after the grant native industrial concerns may obtain a compulsory license to manufacture the patented article. Patent decisions may be appealed from to the general courts. Patent fees are: application fee 25 fl., fees for the patent, second and third years 50 fl. each, and an increasing fee for each subsequent year (a florin is equal to 40.2 cents).

Supplementary References

Motor Fire Engines (*Foreign Review*, August 1912, p. 1241). Complete translation in *Automobile*, August 1 1912.

Use of Preheated Air in Internal-Combustion Engines (*Foreign Review*, April, 1912, p. 615). Translation of the article from the *Gasmotorentechnik*, but without the diagram reproduced in the Foreign Review, in *Automobile*, July 18 1912.

GAS POWER SECTION

PRELIMINARY REPORT OF LITERATURE COMMITTEE

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ARTICLES IN PERIODICALS[1]

AIR TRANSMISSION FOR A GAS-ENGINED VESSEL, "CLOSED CIRCUIT," *The Engineer* (London), July 26, 1912. 3 pp., 7 figs. *cdpA*.

Description and diagram of engine arrangement; one engine coupled to shaft, other connected to shaft engine through heaters and piping and transmitting its power to the shaft engine. Also description of producer.

DIESEL ENGINE, THE LANGEN AND WOLF. *The Engineer* (London), July 26, 1912. 1½ pp., 2 figs. *dmpC*.

Engine made in sizes up to 600 h.p. for auxiliaries on shipboard.

EXPLOSIONS-GASTURBINE, WIRKUNGSGRAD DER, Hans Holzwarth. *Zeitschrift des Vereines deutscher Ingenieure*, June 15-22, 1912. 6¾ pp., 6 tables, 5 curves. *cm*.

Efficiency of the explosion gas turbine. Reply of the author of Die Gasturbine to Professor Stodola's criticism of that book.

GENERATOREN BAUES, NEUES AUS DEM GEBIET DES. *Der praktische Maschinen Konstrukteur*, April 11, 1912, 3 pp., 5 figs. *cpA*.

Data about a producer gas generator with turning grates built by Schneider & Co., Le Creusot, France. Power required to turn grates, 6 h.p.; power required for blower, 4.5 h.p.; air pressure, 40 m/m.

GENERATOR GASFEUERUNG FÜR BRENNÖFEN DER FEINKERAMISCHEN INDUSTRIE, DIE, Walter Stampe. *Braunkohle*, May 24, 1912. 3 figs., 1 curve. *ccpB*.

Discussion over the advantage of using producer gas fired ovens for baking fine ceramic articles.

INTERNAL COMBUSTION ROAD LOCOMOTIVES AND THE DAVIS PARAFFIN CARBURETTER, THE ROYAL AGRICULTURAL SHOW. *The Engineer* (London), July 12, 1912. 2 pp., 5 figs. *cdpB*.

[1] Opinions expressed are those of the reviewer, not of the Society. Articles are classified as *c* comparative; *d* descriptive; *e* experimental; *h* historical; *m* mathematical; *p* practical. A rating is occasionally given by the reviewer, as *A*, *B*, *C*. The first installment was given in The Journal for May 1910.